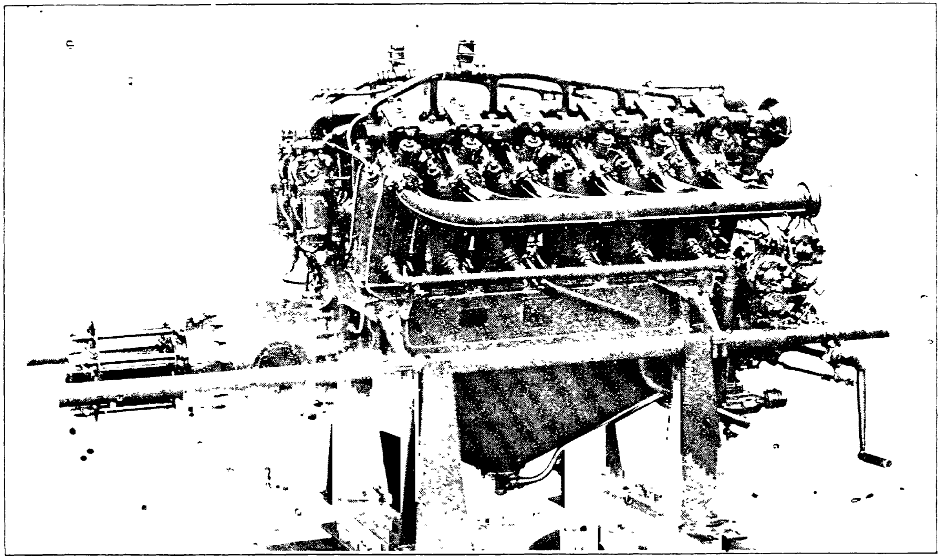
A TEXT-BOOK

ON

HEAT AND HEAT ENGINES.

VOLUME II.

EIGHTEENTH EDITION, REWRITTEN.



360 H P. ROLLS-ROYCE AERO ENGINE

A TEXT-BOOK

ON

HEAT AND HEAT ENGINES

Specially arranged

For the use of Engineers Qualifying for the Institution of Civil Engineers, The Institution of Mechanical Engineers, The Institution of Electrical Engineers, The Diplomas and Degrees of Technical Colleges and Universities, and Honours Certificates of the City and Guilds of London Institute, in Mechanical Engineering, and for Engineers generally.

BY

ANDREW JAMIESON, M.INST.C.E.,

FORMERLY PROFESSOR OF ENGINEERING IN THE ROYAL TECHNICAL COLLEGE, GLASGOW

VOLUME II.

EIGHTEENTH EDITION,

REWRITTEN BY

EWART S. ANDREWS, B.SC.ENG.(LOND.),

FORMERLY DEMONSTRATOR IN THE ENGINEERING LABORATORY OF UNIVERSITY COLLEGE, LONDON,
MEMBER OF COUNCIL OF THE INSTITUTION OF STRUCTURAL ENGINEERS
FELLOW OF THE CHARTERED INSTITUTE OF PATENT AGENTS

With Numerous Diagrams, Folding Plates, and Examination Questions.

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PREFACE

TO VOLUME II.

OF THE EIGHTEENTH EDITION.

THE present volume completes the extension of the late Professor Jamieson's Advanced Text-Book on *Steam and Steam Engines* to cover "Heat and Heat Engines."

With the exception of the lectures dealing with Steam Turbines and Boilers, which are a revised form of the corresponding lectures in Professor Jamieson's seventeenth edition, the whole of the present volume is new, and an attempt has been made to give the student a clear grasp of thermodynamic principles without embarking upon advanced mathematical and metaphysical considerations, and to give sufficient detail description of typical engines to enable the student to have a fair knowledge of the practical application of these principles.

An attempt has been made to enable the student to realise that in practice conditions often obtain which are left out of account in purely theoretical considerations of the subject, and that the word "efficient" is sometimes employed in theoretical considerations without regard to economic conditions, we refer particularly to the fact that the most "efficient" engine from the thermodynamic standpoint is not necessarily the most efficient one from the practical and economic standpoint, which takes into account such questions as prime cost, cost of fuel, freedom from breakdown, and many

other considerations of vital importance in practical life.

While realising this, and thinking it necessary to emphasise it, we do not wish to detract from the great educational and practical value of possessing a clear insight into theoretical principles; we must have these first, so that, later, when we come to study also the practical and commercial conditions, we may form a perspective view of the whole subject, and not be prevented from seeing the wood on account of the trees.

Our thanks are due to the courtesy of the various firms who have been good enough to supply data and illustrations of the engines and accessories associated with their names and described in detail in various parts of the book.

The recent papers of the Associate Membership Examination of the Institution of Civil Engineers have been incorporated by the kind permission of the Council.

EWART S. ANDREWS.

201-206 BANK CHAMBERS,
329 HIGH HOLBORN, LONDON, W.C.,
November, 1922.

ABSTRACT

OF

PREFACE TO THE FIRST EDITION

In a leading article on educational Engineering Treatises, which appeared lately in a well-known journal, the following remarks, amongst others, struck me as being very suggestive to any one engaged in the preparation of a Text-Book for Students, and as well worthy of attention :—

"We are convinced that all the instruction contained in a great number of the engineering books already published, could be printed much more simply and concisely, and also much more lucidly, if authors sought only to impart their knowledge with the greatest brevity, without thinking at all of displaying their own learning or seeking to make a thick volume . . . There is too much paste and scissors work, too much book-making and padding nowadays . . . A considerable number of engineering books are so learned as to be quite over the heads of most students. Many more are so verbose, so laden with abstruse formulæ, letters, and diagrams, that the solution of the simplest question involves hours of time that can ill be spared from other work. It is no doubt true, that many engineering questions demand elaborate writing to give a precise answer with mathematical exactness, but in the majority of engineering practice, absolute exactness of such a nature is not necessary, and if a useful approximation will amply suffice, and is readily obtainable in some simply written book, that is the one that will be adopted."

The object, therefore, aimed at in the following pages, was the production of such a "simply written book" as should *not* be above the heads of my readers, but should bring the information desired, step by step, within their grasp. Whether I have succeeded in accomplishing this object, is a question which, of course, must be decided by those competent to judge.

It is designed to be an easy introduction to Professor Rankine's well-known treatise on *The Steam Engine*, and to Mr. Seaton's practical and highly appreciated *Manual of Marine Engineering*, both issued by the publishers of the present volume.

ANDREW JAMIESON.

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Quantities.	Symbols	Defining Equations.	Practical Units.	Abbreviations of the Practical Units.
FUNDAMENTAL.				
Length, . . .	L, l	...	Yard, . . .	yd
			Foot, . . .	ft.
			Inch, . . .	in
Mass, . . .	M, m	...	Pound, . . .	lb
Time, . . .	T, t	...	Second, . .	s.
			Minute, . .	m.
			Hour, . . .	h
GEOMETRIC.				
Surface, . .	S, s	$S = L^2$	Square foot, . .	sq. ft.
			Square inch, . .	sq. in.
Volume, . . .	V	$V = L^3$	Cubic foot, .	cb ft.
			Cubic inch, . .	cb. in.
Angle, $\angle$ . .	α, β / θ, ϕ	$\alpha = \frac{\text{arc}}{\text{radius}}$	Degree, .	1°
			Minute, . .	$1'$
			Second, . .	$1''$
			Radian $= \frac{180^\circ}{\pi}$.	rn.
MECHANICAL.				
Velocity, . .	v	$v = \frac{L}{T}$	Foot per second, .	$\frac{\text{ft}}{\text{s}}$
Angular velocity, .	ω	$\omega = \frac{v}{L} = \frac{\theta}{t}$	Revs per second, .	r p.s.
			Revs per minute,	r.p.m.
			Radians per second,	ω
Acceleration, . .	a, g	$a = \frac{v}{T}$	Foot per sec. per sec.	$\frac{\text{ft.}}{\text{s}^2}$
Force, . . .	F, f	...	Pound weight (gravitational unit),	lb. wt. (or lb.)
	W, w	$F = M a$	Poundal (absolute unit),	pdl.
Pressure (per unit area),	p	$p = \frac{F}{s}$	Pound per sq. inch,	lb. □″
Work, . . .	$(W h)$	$W h = F L$	Foot-pound, . .	ft.-lb.
Potential energy, .	E_P	$E_P = W h$	Foot-pound, . .	ft.-lb.
Kinetic energy, .	E_K	$E_K = \frac{W v^2}{2 g}$	Foot-pound, . .	ft.-lb.
Power or activity, .	HP	H.P. $= \frac{W h}{T}$	Horse power, .	H.P.
			Ft.-lb. per min., .	ft.-lb./m.
			Ft.-lb. per sec., .	ft.-lb./s.
Moment of inertia, .	I	$I = M k^2$	... .	lb.-ft.^{2}
Density, . . .	ρ	$\rho = \frac{M}{V}$	Pound per cb. ft., .	$\frac{\text{lb.}}{\text{ft.}^3}$
			Pound per cb. in., .	$\frac{\text{lb.}}{\text{in.}^3}$

INDEX TO NOTATION EMPLOYED.

A, a	= Area.
B.H.P.	= Brake horse-power.
B.Th.U.	= British Thermal Unit.
C	= Centigrade.
c_p	= Specific heat at constant pressure
c_v	= Specific heat at constant volume
C	= Constant
D	= Diameter.
d	= Diameter, draught in inches of water.
E_k	= Kinetic energy.
e	= 2·7183 = expotential coefficient
F	= Force; Fahrenheit.
f	= Constant.
g	= Acceleration due to gravity, 32·2 feet per second per second
H, H_1, etc.	= Total heat.
H_i	= Heat at inlet.
H_e	= Heat at exhaust.
h	= Heat; height
H.P.	= Horse-power.
I.H.P.	= Indicated horse-power.
J	= Joule's equivalent.
Kw.	= Kilowatt.
k	= Constant, specific heat of superheated steam.
L, L_1, etc	= Latent heat, length.
L_i	= Latent heat at inlet.
L_e	= Latent heat at exhaust.
l	= Length
N	= Revolutions per minute.
P, P_1, etc.	= Pressure
P_c	= Coefficient of performance.
p, p_1, etc.	= Pressure, pitch.
q	= Dryness coefficient.
R_η	= Rankine efficiency coefficient.
r	= Ratio of expansion.
r_p	= Pressure ratio.
S, S_1, etc.	= Sensible heat.
S_i	= Sensible heat at inlet.
S_e	= Sensible heat at exhaust.
S_s	= Shear strength

S_t	= Tensile strength.
s, s_1, etc.	= Sensible heat, specific heat.
t, t_1, etc.	= Temperature, thickness.
t_s	= Temperature of superheat.
V	= Volume, unit volume of saturated steam.
v	= Volume, velocity
W	= Weight.
w	= Unit volume of water
x, x_1, etc.	= Dryness coefficient.
α	= Angle.
β	= Angle.
γ	= $c_p \div c_v$ = ratio of specific heats of a gas.
η	= Efficiency.
η_a	= Air standard efficiency.
η_v	= Efficiency of constant volume cycle using the variable specific heat theory.
π	= 3·1416.
τ, τ_1	= Absolute temperature.
τ_i	= Absolute temperature at inlet.
τ_e	= Absolute temperature at exhaust.
φ	= Entropy.
φ_l	= Entropy of liquid
φ_v	= Entropy of vapour.
φ	= $\varphi_l + \varphi_v$ = total entropy.

HEAT AND HEAT ENGINES

VOL. II.

LECTURE I.

ELEMENTARY THERMODYNAMICS.

CONTENTS.—First Law of Thermodynamics—Second Law of Thermodynamics—Indicated and Brake Efficiencies of Engines—Laws of Expansion of Perfect Gases—Work done in Isothermal Expansion—Work done in Expansion when PV^n Constant—Graphical Construction for Expansion PV^n Constant—Carnot's Cycle—Adiabatic Expansion—Efficiency in Carnot's Cycle—Reversible Cycles—A Reversible Cycle has the maximum possible Theoretical Efficiency

THERMODYNAMICS is a subject which engineering students often find difficult to understand; this is probably due to the fact that it is built upon a foundation of theories which many lecturers have set out as fixed laws, whereas they are in reality only assumptions which, as far as we are able to say at present, are reasonable If these assumptions are correct, we are able to deduce certain results from them, these results being called the laws of thermodynamics The engineer's aim should be to follow the reasoning and to apply the results to practical engineering problems; the real test as to whether an engineer understands the theory of heat engines is whether he can use it to improve the design of engines, but the development of his imagination by the introduction of abstract conceptions that assist in the logical development of the theory is a matter of considerable value which the practical engineer is apt sometimes to forget.

First Law of Thermodynamics. *When heat energy is converted into mechanical energy a definite quantity of heat goes out of existence for every unit of work done; and conversely when mechanical energy is converted into heat energy a definite quantity of heat comes into existence for every unit of work expended.*

We have considered this law already,* and have seen that

* See Lecture VI., Vol I

according to the best experimental evidence available a British thermal unit (pound-degree-Fahrenheit) is equivalent to 778 foot-pounds; a Centigrade heat unit (pound-degree-Centigrade) is equivalent to 1,400 foot-pounds; a kilogramme-degree-Centigrade is equivalent to 427 kilogramme-metres. These quantities of their respective units are often referred to as *Joule's Equivalent.*

Second Law of Thermodynamics. *It is impossible for a self-acting machine, unaided by any external agency to convey heat from one body to another at a higher temperature.* This is the form of the law given by Clausius; the law has been from the outset a matter of great controversy, a very valuable and interesting account of which is to be found in Silvanus Thompson's *Life of Lord Kelvin*. It does not present in this form much obvious practical meaning, but, as we will show later, the following very important rule can be deducted therefrom :—

If a heat engine has an absolute inlet temperature τ_i *and an exhaust or outlet temperature* τ_e, *the maximum possible efficiency of the engine, even if it is thermally and mechanically perfect, will be given by*

$$\textit{Maximum efficiency} = \eta_{max} = \frac{\tau_i - \tau_e}{\tau_i}.$$

The efficiency of an engine is defined by

$$\text{Efficiency} = \frac{\text{Heat converted into work}}{\text{Heat taken in by the engine}}.$$

Indicated and Brake Efficiencies of Engines. In dealing with efficiencies of heat engines we have to distinguish between the efficiency estimated from the indicated horse-power and that from the brake horse-power; these we will refer to as the *indicated efficiency* and *brake efficiency* respectively. The indicated efficiency is of interest as indicating the proportion of the energy converted into work in the engine cylinder, but the brake efficiency is of greater practical importance, since it gives us the proportion of the energy actually given out by the engine. In the case of turbines, we are unable to measure the indicated efficiency. The *mechanical efficiency* of an engine is given by

$$\text{Mechanical efficiency} = \frac{\text{Brake horse-power}}{\text{Indicated horse-power}}$$

$$= \frac{\text{Brake efficiency}}{\text{Indicated efficiency}}.$$

The highest indicated efficiency which has been obtained in a steam engine using saturated steam is about 20 per cent., the corresponding brake efficiency being about 18 per cent., and using superheated steam an indicated efficiency of 37 per cent. has been obtained, it should be remembered that these are the highest results obtained under the best conditions, and that in most cases in practice the efficiencies obtained will be much less; moreover, the loss of heat in the boiler and super-heater are not taken into account About 18 per cent represents also the highest brake efficiency obtained with steam turbines

The highest known indicated efficiency for any heat engine is about 41 per cent, and was obtained by the Diesel engine, the mechanical efficiency of this engine is, however, less than that of the best steam engines, being about 77 per cent, so that the brake efficiency obtained is about 31 per cent.

Scientific and Commercial Efficiency Before leaving this general consideration of efficiency, we have to remember that this efficiency is only in the scientific sense Before we can deal with commercial efficiency, which after all, is the final criterion in practice, we have to consider the relative costs of fuel for various kinds of engines, and in comparing, say, coal-heated steam and oil engines we should express our results in pounds of fuel per H P-hour, and then find the cost of fuel at the locality in question. Secondary conditions, such as space and upkeep, have an important bearing upon the practical utility and, therefore, the real efficiency, for the present, however, we will restrict ourselves to the theoretical efficiency obtainable under the best conditions.

Numerical Example. In a test of a steam engine using steam at 141 lbs per square inch superheated to 420° C., 9 lbs of steam were used per I H.P.-hour, the feed-water temperature being 24° C., and the exhaust temperature 50° C Find the highest efficiency possible with this engine and compare it with that actually obtained.

In this case we have

$$\eta = \frac{\tau_i - \tau_e}{\tau} = \frac{420 - 50}{420 + 273} = \frac{370}{693} = 53{\cdot}4 \text{ per cent.}$$

The total heat of saturated steam per pound at 141 lbs. per square inch is 667 in Centigrade heat units, the saturation temperature being 183° C; the superheat is, therefore, 420 − 183

= 237° C., so that, taking the specific heat of superheated steam as ·52, we have—

$$\text{Total heat of superheated steam} = 667 + \cdot52 \times 237$$
$$= 790 \text{ C.H.U.}$$

$$\text{Heat given up per pound of steam} = 790 - 24 = 766 \text{ C.H.U.}$$

$$\text{Heat given up per H.P.-hour} = 766 \times 9 = 6{,}890.$$

$$\text{Indicated efficiency} = \frac{33{,}000 \times 60}{6{,}890 \times 778}$$
$$= 37 \text{ per cent.}$$

Laws of Expansion of Perfect Gases.—In order to study this second law of thermodynamics with reference to certain standard theoretical engine cycles, we will remind ourselves of the laws of perfect gases which were dealt with in Lectures XII. and XIII. of Volume I., and deduce certain additional useful formulæ from them.

By a combination of what are commonly referred to as Boyle's and Charles' laws, we have

$$\frac{\text{P V}}{\tau} = \text{constant} = c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

If P is in lbs. per square foot, V in cubic feet, and τ in ° F. absolute, $c = 53\cdot2$ for air.

Also, $$c = \frac{C_p - C_v}{J} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where C_p = capacity for heat or specific heat of a gas at constant pressure,

C_v = capacity for heat or specific heat of a gas at constant volume,

J = Joule's equivalent.

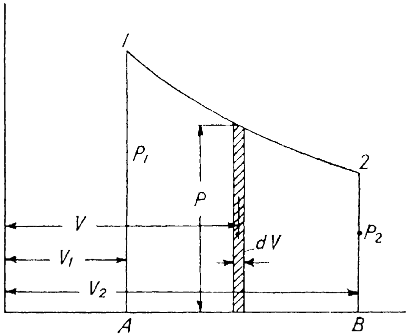
The ratio $\frac{C_p}{C_v}$ enters into many of our equations, and is given the symbol γ. For air and several other gases we may take $\gamma = 1\cdot41$; values for many gases are given in the Appendix.

We will here note that the specific heats of gases vary with the temperature; this has an important bearing upon the advanced theory of internal combustion engines.

Work done in Isothermal Expansion (Temperature Constant).—We have seen already that the work done in expanding from pressure P_1 and volume V_1 to pressure P_2 and volume V_2 is given by the area 1 2 B A of the corresponding P V diagram. Considering a narrow strip of this diagram, we have—

$$\text{Area of strip} = P\,dV = \frac{c\,\tau\,dV}{V} \text{ (by 1).}$$

$$\therefore \text{ Total area from A to B} = \int_{V_1}^{V_2} \frac{c\,\tau\,dV}{V}.$$



If the expansion is isothermal, τ is constant and c is constant, so that we say—

$$\text{Total area from A to B} = c\,\tau \int_{V_1}^{V_2} \frac{dV}{V}$$

$$= c\,\tau\,(\log_e V_2 - \log_e V_1)$$

$$= c\,\tau \log_e \frac{V_2}{V_1}, \qquad \qquad (3)$$

i.e., $$\text{Work done} = c\,\tau \log_e r, \qquad \qquad (3a)$$

where r is the ratio of expansion $\frac{V_2}{V_1}$.

Work Done in Expansion when $P V^n$ = Constant.—In this case we have, as before

$$\text{Area of strip} = P\, d\, V.$$

But we may write

$$P\, V^n = P_1\, V_1^n = P_2\, V_2^n.$$

$$\therefore \quad \text{Area of strip} = \frac{P_1\, V_1^n\, d\, V}{V^n}.$$

$\therefore$ Total area from A to B

$$= \int_{V_1}^{V_2} \frac{P_1\, V_1^n\, d\, V}{V^n}$$

$$= P_1\, V_1^n \int_{V_1}^{V_2} \frac{d\, V}{V^n}$$

$$= \frac{P_1\, V_1^n\, (V_2^{1-n} - V_1^{1-n})}{(1-n)}$$

$$= P_1\, V_1 \frac{\left\{1 - \left(\frac{V_2}{V_1}\right)^{1-n}\right\}}{n-1} \quad . \quad . \quad (4)$$

$$\text{Work done} = \frac{P_1\, V_1\, (1 - r^{1-n})}{n-1}, \quad . \quad . \quad . \quad (5)$$

but since $P_1\, V_1^n = P_2\, V_2^n$, we see that

$$P_1\, V_1 \left(\frac{V_2}{V_1}\right)^{1-n} = P_2\, V_2,$$

i.e., $$\text{Work done} = \frac{P_1\, V_1 - P_2\, V_2}{(n-1)}. \quad . \quad . \quad (6)$$

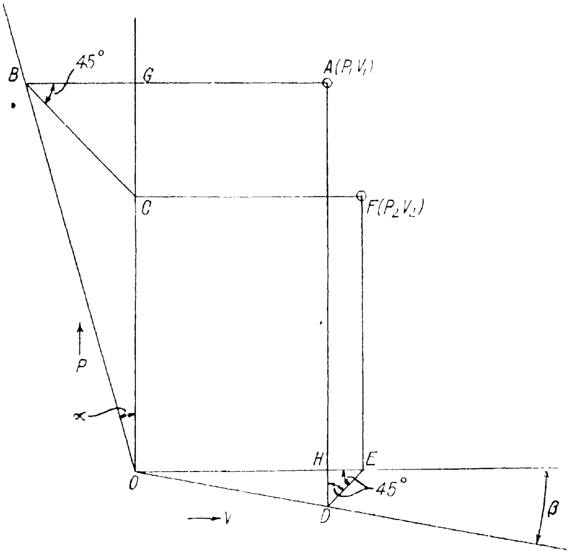
Also, if the gas is perfect, we have

$$P_1\, V_1 = c\, \tau_1 \text{ and } P_2\, V_2 = c\, \tau_2.$$

$$\therefore \quad \text{Work done} = \frac{c\,(\tau_1 - \tau_2)}{(n-1)}. \quad . \quad . \quad . \quad . \quad (7)$$

Graphical Construction for Expansion Curve PV^n = Constant.—Let A be any given point on the curve. Draw a line O B at any convenient angle α to the pressure axis, and draw a line at angle β to the volume axis such that

$$\tan\beta = \frac{1-(1-\tan\alpha)^n}{(1-\tan\alpha)^n} \qquad (a)$$



Draw a horizontal line through A to meet O B in B, and draw B C at 45°, and draw a horizontal line C F through C; then draw A D vertically and draw D E at 45°, then projecting E F vertically to intersect C F in F.

Then F is a point on the required curve.

Proof.—Tan $\alpha = \frac{BG}{OC} = \frac{CG}{OG}$.

$$\therefore \quad 1 - \tan\alpha = 1 - \frac{\mathrm{CG}}{\mathrm{OG}} = \frac{\mathrm{OG} - \mathrm{CG}}{\mathrm{OG}} = \frac{\mathrm{OC}}{\mathrm{OG}} = \frac{\mathrm{P}_2}{\mathrm{P}_1}. \quad . \quad (b)$$

$$\tan\beta = \frac{\mathrm{HD}}{\mathrm{OH}} = \frac{\mathrm{HE}}{\mathrm{OH}} = \frac{\mathrm{V}_2 - \mathrm{V}_1}{\mathrm{V}_1}.$$

$$\therefore \quad \frac{\mathrm{V}_2 - \mathrm{V}_1}{\mathrm{V}_1} = \frac{\mathrm{V}_2}{\mathrm{V}_1} - 1 = \frac{1 - (1 - \tan\alpha)^{\frac{1}{n}}}{(1 - \tan\alpha)^{\frac{1}{n}}}$$

$$= \frac{1}{(1 - \tan\alpha)^{\frac{1}{n}}} - 1$$

$$\therefore \quad \frac{\mathrm{V}_2}{\mathrm{V}_1} = \frac{1}{(1 - \tan\alpha)^{\frac{1}{n}}}.$$

$$\therefore \quad \left(\frac{\mathrm{V}_2}{\mathrm{V}_1}\right)^n = \frac{1}{(1 - \tan\alpha)} = \frac{\mathrm{P}_1}{\mathrm{P}_2} \text{ [from } (a)\text{]}.$$

$$\therefore \quad \mathrm{P}_1 \mathrm{V}_1{}^n = \mathrm{P}_2 \mathrm{V}_2{}^n.$$

Carnot's Cycle. Carnot considered an imaginary engine in which a perfect gas passed through a number of stages, the complete sequence of which is termed a cycle.

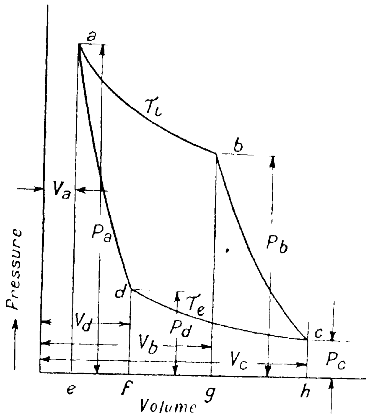
In thermodynamics we speak of the *working substance;* this is the general name for the material which takes in and rejects heat, doing work in the process. In practice the principal working substances which we have to consider are steam, coal gas, and mixtures of oil or petrol vapour and air, but theoretically the substance might be solid. We might, for instance, have an imaginary heat engine consisting of a long metal rod provided with a pawl acting upon the ratchet wheel, if the rod were heated the pawl would drive the wheel forward and would return idly on the return stroke when the rod was cooled. Thus by alternate heating and cooling of the rod a continuous rotation of the wheel could theoretically be obtained.

In Carnot's imaginary engine the working substance is a perfect gas, and the piston and cylinder are composed of a material which is a perfect non-conductor of heat, except at the end of the cylinder which is composed of a well-conducting material.

Suppose that we have two bodies, X and Z, which are sources of heat maintained at absolute temperatures τ_i and τ_e respectively, and a non-conducting cover Y, the bodies X, Y, Z being such that they can be selectively placed in contact with the bottom of the cylinder.

Suppose further that the cylinder contains a unit weight of gas, the volume and pressure of which at an absolute temperature τ_i are V_a and P_a (see figure), and that the following cycle of operations is then performed :—

(1) Apply body X to end of the cylinder and allow gas to expand, thus driving the cylinder forward until the pressure



is P_b and volume V_b. During this expansion the temperature remains constant at τ_i, so that the *expansion is isothermal*.

(2) Place the non-conducting body Y in contact with the end of the cylinder and allow the gas to continue to expand until the pressure is P_c and volume V_c. During this expansion no heat can enter or leave the cylinder; the *expansion is said, therefore, to be adiabatic*, and since the gas has done work in expanding, such work must have been taken out of the internal energy of the gas, so that the temperature will have fallen. We imagine that the expansion has been continued until the temperature has fallen to τ_e.

(3) Place the body Z in contact with the end of the cylinder,

and drive the piston back until a point d is reached on the P V diagram (or "indicator diagram," as it is commonly called), at which the pressure is P_d and the volume V_d. During this stage the temperature remains constant at τ_e, so that the gas is *compressed isothermally*. The point d is so chosen that the fourth stage will complete the cycle.

(4) Replace the non-conducting body Y against the end of the cylinder and continue the compression. It is clear that in this stage we have *adiabatic compression*, and since work is done on the gas, without any heat being allowed to escape, its internal energy and, therefore, its temperature must rise. The point d will have been so chosen that when the volume is V_a the pressure will be P_a, the gas being thus returned to its initial condition.

Adiabatic Expansion. Before proceeding further to deduce the theoretical efficiency obtainable from Carnot's cycle, we will consider the relation which must occur between pressure and volume in order that expansion may be adiabatic—*i.e.*, without increase or decrease of heat from or to an outside source.

Joule proved experimentally that if a gas were perfect the internal energy would depend only upon its absolute temperature, and not upon the manner in which its pressure and volume had changed in reaching that temperature.

In any operation of heating a substance, we must have

Work equivalent of heat added

= external work done + increase of internal energy

Now, let a unit mass of gas at absolute temperature τ_1 be heated until its temperature $= \tau_2$, and let the volume remain constant so that the external work done is zero.

Then heat added $= C_v (\tau_2 - \tau_1)$

$\therefore$ Since external work done $= 0$,

$$\text{Increase of internal energy} = \frac{C_v}{J}(\tau_2 - \tau_1). \quad . \quad . \quad (8)$$

Now, in view of Joule's law, this must be the measure of change of internal heat energy in any operation by which the temperature changes from τ_2 to τ_1.

In the case of *adiabatic expansion*, work equivalent of heat added $= 0$.

$$\therefore \quad \text{External work done} = \text{decrease of internal energy}$$
$$= \frac{C_v}{J}(\tau_1 - \tau_2).$$

∴ By equation (7), if the expansion follows the law $P V^n$ = constant—

$$\frac{c}{(n-1)} = \frac{C_v}{J}. \qquad (9)$$

and ∴ by equation (2)

$$\frac{C_p - C_v}{J(n-1)} = \frac{C_v}{J},$$

i.e.,
$$\left(\frac{C_p - C_v}{C_v}\right) = (n-1)$$

$$\left(\frac{C_p}{C_v} - 1\right) = (n-1),$$

i.e.,
$$(\gamma' - 1) = (n-1)$$

or
$$\gamma' = n$$

∴ In adiabatic expansion, $P V^\gamma$ = Constant.

This enables us to write equations (5), (6), and (7) in the following forms :—

Work done in adiabatic expansion $= \dfrac{P_1 V_1 (1 - r^{1-\gamma})}{\gamma' - 1}$ (5a)

$$= \frac{P_1 V_1 - P_2 V_2}{\gamma' - 1} \qquad (6a)$$

$$= \frac{c(\tau_1 - \tau_2)}{\gamma - 1} \qquad (7a)$$

We also obtain the following relation between temperatures and volumes in adiabatic expansion :—

Since $\dfrac{P_1 V_1}{\tau_1} = \dfrac{P_2 V_2}{\tau_2}$ and $\dfrac{P_1}{P_2} = \left(\dfrac{V_2}{V_1}\right)^\gamma$

$$\frac{\tau_1}{\tau_2} = \frac{P_1 V_1}{P_2 V_2} = \left(\frac{V_2}{V_1}\right)^\gamma \cdot \frac{V_1}{V_2} = \left(\frac{V_2}{V_1}\right)^{\gamma - 1} \qquad (10)$$

Efficiency in Carnot's Cycle.—Referring back to the diagram on p. 9 for Carnot's cycle, we have—

Work done = area $a\,b\,g\,e$ + area $g\,h\,c\,h$ — area $f\,d\,c\,h$
— area $e\,a\,d\,f$,

but area $g\,b\,c\,h$ = area $e\,a\,d\,f = \dfrac{c\,(\tau_i - \tau_e)}{\gamma - 1}$.

$$\therefore \text{ Work done} = \text{area } a\,b\,g\,e - \text{area } f\,d\,c\,h$$

$$= [\text{from eq. (3)}]\; c\,\tau_i \log_e \frac{V_b}{V_a} - c\,\tau_e \log_e \frac{V_c}{V_d}; \qquad (11)$$

but $$\frac{\tau_i}{\tau_e} = \left(\frac{V_c}{V_b}\right)^{\gamma-1} = \left(\frac{V_d}{V_a}\right)^{\gamma-1}$$

$$\therefore \quad \frac{V_c}{V_b} = \frac{V_d}{V_a}.$$

$$\therefore \quad \frac{V_b}{V_a} = \frac{V_c}{V_d}.$$

$\therefore$ (11) becomes

$$\text{Work done} = c\,(\tau_i - \tau_e) \log_e \frac{V_b}{V_a}. \qquad (12)$$

Now, work taken in while in contact with the source of heat X

$$= \text{area } a\,b\,g\,e = c\,\tau_i \log_e \frac{V_b}{V_a}. \qquad (13)$$

$\therefore$ **Efficiency of cycle** $= \dfrac{\text{Work done}}{\text{Work taken in}}$

$$= \frac{c\,(\tau_i - \tau_e) \log_e \dfrac{V_b}{V_a}}{c\,\tau_i \log_e \dfrac{V_b}{V_a}}$$

$$= \frac{(\tau_i - \tau_e)}{\tau_i}. \qquad (14)$$

Now, since τ_i is the inlet temperature and τ_e is the outlet temperature, we see that the theoretical efficiency of Carnot's cycle is the same as that which we stated on p. 2 to be the maximum efficiency obtainable in any heat engine.

Reversible Cycles.—Now, suppose that we were to reverse Carnot's cycle—*i.e.*, if we force the imaginary engine to act so that the work diagram is traced out in the opposite direction. The following cycle would then occur:

(1) Starting with an amount of working substance V_a in the cylinder at temperature τ_i in contact with the non-conducting body Y we allow it to expand until its temperature is τ_e.

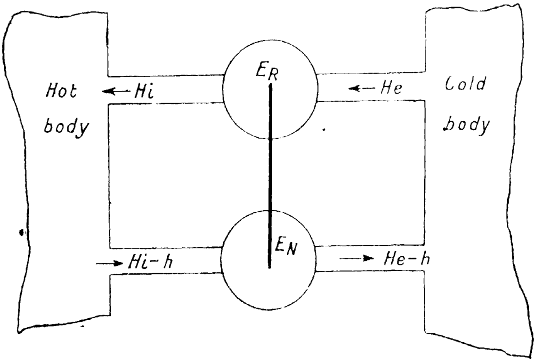
(2) We then place the body Z in contact with the cylinder and allow the piston to move out until the volume is V_d.

(3) Next place the non-conducting body Y in contact with the cylinder and compress the gas adiabatically until the temperature is τ_i.

(4) Place the body X in contact with the cylinder, and continue the compression until the volume V_a is again reached. During this stage, heat will flow into the body X.

Our engine now acts as a heat pump, and it is clear from the diagram that we have transferred from the cold body Z to the hot body X the same amount of heat as we took from the hot body in the previous case, and that we have expended the same amount of work as we converted into use in the previous case.

A heat engine in which this is possible is called a thermodynamically reversible engine, and the cycle is called a *reversible cycle*.



A Reversible Cycle has the Maximum Possible Theoretical Efficiency.—We will next prove that a reversible cycle is the most efficient cycle possible; we have shown that in one such cycle—viz. Carnot's—that the theoretical efficiency $= \frac{\tau_1 - \tau_2}{\tau_1}$; we shall, therefore, be able to state that.

The theoretical efficiency of any reversible cycle with absolute inlet temperature τ_i and exhaust temperature τ_e is equal to $\frac{\tau_i - \tau_e}{\tau_e}$.

Proof. Suppose that we have two engines E_R and E_N, the former being reversible and the latter not; and suppose that E_N is more efficient than E_R, and suppose that the reversible engine in each cycle takes in its normal working an amount H_i from the hot body and gives up an amount H_e to the cold body, doing an amount of work $(H_i - H_e)$.

Then if E_N is more efficient than E_R, it will require less heat — say $(H_i - h)$ — to do the same amount of work.

Now, let the two engines be coupled together so that E_N drives E_R without any frictional loss. Then since E_R is reversible, it will pump an amount of heat H_e from the cold body and deliver an amount of heat H_i to the hot body, while E_N takes $H_i - h$ from the hot body and delivers $H_e - h$ to the cold body.

In each cycle, therefore, E_R will pump into the hot body more heat than E_N takes from it, and since one engine drives the other, no external energy is used. We shall, therefore, be continuously transferring heat from the cold body to the hot body without the expenditure of work, and *this is contrary to the second law of thermodynamics*.

We, therefore, see that it is impossible for a non-reversible engine to be more efficient than a reversible one.

Summary of Argument on Reversible Cycles.—We will now collect together the various statements that we have made or proved.

(1) If the second law of thermodynamics is true, a reversible cycle is the most efficient cycle possible for a heat engine.

(2) In one reversible cycle—viz., Carnot's imaginary cycle—the thermal efficiency can theoretically be equal to

$$\eta_t = \frac{\tau_i - \tau_0}{\tau_i},$$

where τ_i = absolute inlet temperature,

τ_0 = absolute outlet temperature.

(3) In every reversible cycle, therefore, the highest possible thermal efficiency is given by the above equation.

LECTURE I.—QUESTIONS.

1. Find the absolute zero of temperature from the following data:—Under a pressure of 2,116·4 lbs. per square foot and at 32° F., the volume of 1 lb. of air is 12·387 cubic feet, while at 104° F. it is 14·2 cubic feet. *Ans.* − 461° F.

2. If 1 lb. of air does 390·6 ft.-lbs. of work without receiving or rejecting any heat, what will be its fall in temperature? *Ans.* 3° F.

3. A quantity of gas occupying $6\frac{1}{2}$ cubic feet at a temperature 60° F. is compressed isothermally to $\frac{1}{3}$ of its volume. It is then cooled at constant pressure. Find the volume of the gas when the temperature has been lowered in this way to 32° F. *Ans.* 2·05 cubic feet.

4. One lb. of air is compressed to two atmospheres pressure, the temperature being 20° C.; what is its volume? It receives heat energy equivalent to 1,000 ft.-lbs., its volume remaining constant; find its new pressure and temperature. The specific heat of air at constant pressure is 0·238. *Ans.* 6·675 cubic feet, 1·15 atmospheres, 26° C.

5. Gas is compressed adiabatically to $\frac{1}{10}$ of its volume. If the original temperature is 16° C., and $\gamma = 1·4$, find the final temperature. *Ans.* 455° C.

6. What is meant by the term "reversible" as applied to a heat engine; prove that when working between two given temperatures no engine can be more efficient than a reversible engine. What is the highest possible efficiency of an engine with inlet temperature 325° F. and outlet temperature 126° F. *Ans.* 25·3 per cent.

7. If a steam engine is worked between 320° F. and 100° F., and the ratio of its efficiency to that on the Carnot cycle is ·55, how many B.Th.U. would it require per horse-power per minute? *Ans.* 274 B.Th.U.

8. Compressed air at 120 lbs. per square inch absolute is drawn into a cylinder, and is then expanded to 6 times its original volume. Determine the mean absolute pressure (*a*) if the temperature is constant, (*b*) if the cylinder is non-conducting and $\gamma = 1·408$. If the initial temperature is 70° F., find the final temperature in the latter case. *Ans.* (*a*) 55·8 lbs. per square inch; (*b*) 45·4 lbs. per square inch; final temperature, − 205·4° F.

9. An engine uses 10 lbs. of steam per minute, the feed temperature is 60° F., the boiler temperature 300° F., and that of the condenser 104° F., what is the theoretical maximum efficiency of the engine? Find how many heat units have been used per minute and what horse-power would be developed if the engine were a perfect one. *Ans.* 25·8 per cent.; 11,400 B.Th.U.; 69 H.P.

10. The volume of 1 lb. of air at 0° C. and 1 atmosphere pressure is 12·4 cubic feet; what will it be at $2\frac{1}{2}$ atmospheres and 130° C.? If it receives heat energy equivalent to 300,000 foot-lbs. at constant volume, what will be the new temperature and pressure, given that the specific heat of air at constant pressure is 0·238? *Ans.* 7·2 cubic feet; 7·7 atmospheres; 831° C. absolute.

LECTURE I. A.M.INST.C.E. QUESTIONS.

1. Define the following terms :—Specific heat of a gas at constant volume, adiabatic expansion, latent heat of evaporation, total heat of a saturated vapour; heat of a liquid. State the relation existing between the last three terms for the same liquid.

2. What is meant by a reversible cycle and by an irreversible cycle? Which portions of the cycle of an actual simple steam engine are irreversible?

3. One pound of air is compressed adiabatically to one-fourth of its original volume. Calculate the temperature reached, taking the initial temperature at 60° F. and $\gamma = 1.4$. How many B.Th.U. must be removed from the compressed air in order that it may be cooled to the original temperature, without changing its compressed volume? $C_v = 0.17$.

4. Obtain the usual expression for the adiabatic expansion of a perfect gas.

5. What is meant, in thermodynamics, by the terms "reversible" and "irreversible" operations? Show that, within the same temperature limits, no engine can be more efficient than a reversible engine.

6. State the first and second laws of thermodynamics, and prove by their means that only a percentage of the heat supplied to a heat engine can be converted into work.

7. How would you calculate the heat expended in external work during the expansion of a given volume of a perfect gas according to the law $PV^n = $ constant, if the initial and final temperatures of the gas, the value of the constant, and the specific heats at constant pressure and constant volume were known? State the equations you would use, and show how these are derived from the fundamental properties of perfect gases.

8. What is meant by a reversible heat engine? Find an expression for the efficiency of a heat engine working on the Carnot cycle.

9. Show that, for a perfect gas, the difference between the specific heats at constant pressure and constant volume is equal to the work done in raising the gas 1° in temperature. Hence show that, if the absolute temperature and pressure of a gas are known, its volume is also known.

LECTURE II.

ENTROPY.

CONTENTS.—Definition of Entropy—Entropy and Adiabatic Expansion—Temperature-Entropy Diagrams—Carnot's Cycle from the Entropy Standpoint—Temperature-Entropy Diagram for Steam—Units of Entropy—The Rankine—Dryness after Adiabatic Expansion—Entropy Diagram for Superheated Steam—Constant Volume Lines on the Entropy Diagram for Steam—Numerical Examples—Clapeyron's Equation for the Relation between Volume and Pressure of Saturated Steam—Questions.

WE come now to consider a thermal quantity, to which the name *entropy* has been given, which has been the cause of great controversy among exponents of thermodynamics, and which usually presents considerable difficulty to practical engineers.

Before attempting to define the quantity, we will point out that one, if not the principal difficulty in the matter is that entropy is an abstract conception that we cannot express in very familiar units; in this respect it bears some resemblance to moment of inertia. We shall probably find that we shall do well to attempt to understand some of the properties of entropy rather than try to understand completely the definition of it.

Definition of Entropy.—*If a small quantity of heat* δH *is added to or taken from a body whose absolute temperature is* τ, *the quantity* $\frac{\delta H}{\tau}$ *is called the change of entropy of the body.*

If we wish to obtain a measure of the total entropy possessed by a body in a given condition, we must first agree upon a zero or starting-point. We will follow the usual practice of taking the freezing point of water, viz., 0° C. or 32° F., as the starting point.

If we now divide the stages by which the substance has attained its given condition into a large number of parts, over each of which the temperature may be regarded as constant, and divide the change of heat in each stage by the corresponding absolute temperature, we can add the separate results together to obtain a measure of the entropy of the substance in the given condition.

The symbol φ is used to indicate entropy, so that we can express in mathematical language the addition of separate parts referred to above as

$$\varphi = \Sigma \frac{\delta H}{\tau} \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

or

$$\delta \varphi = \frac{\delta H}{\tau} \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Entropy and Adiabatic Expansion. We have already seen that in adiabatic expansion of a substance there is no change in the quantity of heat contained in the substance; it follows, therefore, that in adiabatic expansion there is no change of entropy. For this reason some writers have called adiabatic expansion *isentropic* expansion; others have used this result as a foundation for their definition of entropy by stating that "entropy is that thermal property of a substance which remains constant in adiabatic expansion." This definition is not one that we recommend, since entropy is not the only quantity which remains constant during adiabatic expansion. We might as well define the temperature of a liquid as that quantity which remains constant while the liquid boils.

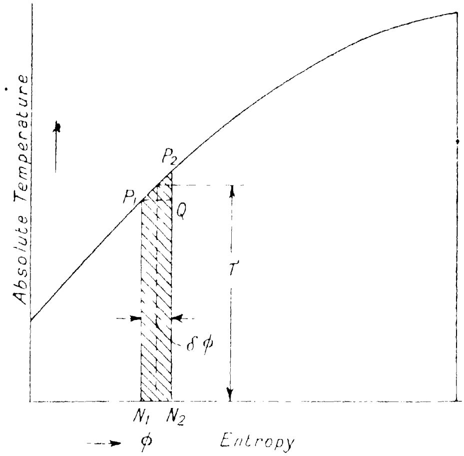
Temperature-Entropy Diagrams. Suppose that we plot a diagram with absolute temperatures as ordinates and entropy as abscissæ, as shown on the accompanying figure, and we take two points, P_1 P_2, very near to each other on the diagram. Then, if we draw ordinates $P_1 N_1$, $P_2 N_2$, we see that $N_1 N_2$ = change of entropy between P_1 and P_2 = $\delta \varphi$; now the area of this shaded strip = $N_1 N_2 \left(\frac{P_1 N_1 + P_2 N_2}{2}\right) = \delta \varphi \times \tau$, if P_1 and P_2 are sufficiently close together for the change of temperature to be negligible compared with either temperature.

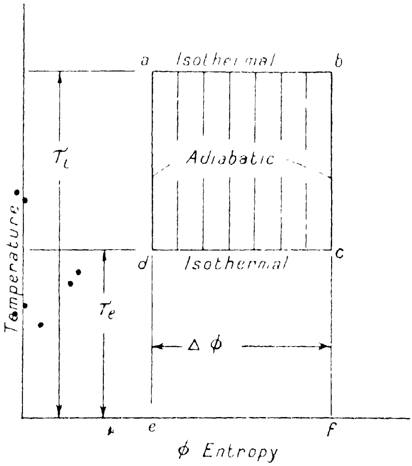
$\therefore$ Area of strip = $\delta \varphi \times \tau = \delta H$ = change of heat.

Since we may divide up the whole diagram in this manner, and the area of each strip will represent the corresponding change of heat, we see that *the area of an entropy diagram represents change of heat—that is, quantity of heat gained or lost.*

Carnot's Cycle. Now let us draw the temperature-entropy ($\tau \varphi$) diagram for an imaginary engine working upon Carnot's cycle (see p. 2).

It is clear that since temperature remains constant in isothermal expansion, the corresponding portion of the $\tau \varphi$ diagram will


Absolute Temperature
P_2
P_1
Q
T
$\delta\phi$
N_1 N_2
ϕ Entropy


a Isothermal b
T_i
Adiabatic
Temperature
d Isothermal c
T_e
$\Delta\phi$
e f
ϕ Entropy

be a horizontal straight line; also that corresponding to adiabatic expansion, during which entropy is constant, will be a vertical straight line. The student should have no difficulty in following these statements if he refers to the accompanying diagram.

During the first stage, therefore (isothermal expansion), of Carnot's cycle, the $\tau\varphi$ diagram passes horizontally from a to b; during the next stage (adiabatic expansion) it passes vertically from b to c, during the third (isothermal compression) the diagram passes horizontally from c to d, and the diagram is completed by the vertical $d\,a$ corresponding to the final adiabatic compression.

From a study of the diagram we see that

Work or energy put in = area $a\,b\,f\,e$ = $\tau_1 \times \triangle\varphi$.
,, returned = area $c\,f\,e\,d$ = $\tau_e \times \triangle\varphi$.
,, done by engine = area $a\,b\,c\,d$ = $(\tau_1 - \tau_e)\triangle\varphi$.

$$\therefore \qquad \eta = \text{Efficiency of imaginary engine} = \frac{\text{Work done}}{\text{work put in}}$$

$$= \frac{(\tau_1 - \tau_e)\triangle\varphi}{\tau_1 \triangle\varphi}.$$

$$i.e., \qquad \eta = \frac{\tau_1 - \tau_e}{\tau_1} \qquad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \qquad (2)$$

This agrees with the value which we obtained previously.

Temperature-Entropy ($\tau\varphi$) **Diagram for Steam.**—Suppose that we start with a pound of water at freezing point, the temperature of which on the absolute scale we will call τ_0, and generate steam at a temperature τ_1, the pressure remaining constant. We have already seen in Volume I that the temperature of the water will rise gradually to τ_1, the heat that has been added being called *sensible heat*, and that the temperature will then remain constant until we have added a further amount of heat equal to the *Latent Heat* L_1 at τ_1, we then have a pound of steam at temperature τ_1.

The increase of entropy during this period may be considered as made up of two parts, viz.:—

(1) Increase of entropy of water = $\triangle\varphi_1$.
(2) ,, in changing water to steam = $\triangle\varphi_2$

$$\text{Now } \triangle\varphi_1 = \sum_{\tau_0}^{\tau_1} \frac{\delta Q}{\tau}.$$

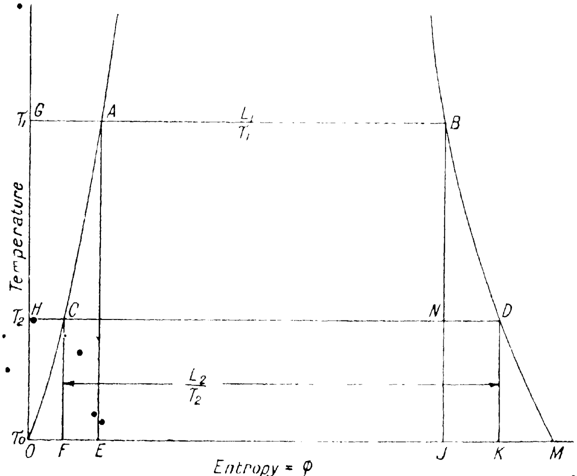
If the specific heat of water is constant and equal to 1, since δQ = specific heat $\times\ \delta\,\tau$, we see that

$$\triangle\,\varphi_1 = \sum_{\tau_0}^{\tau_1} \frac{\delta\,\tau}{\tau}.$$

If this summation be effected mathematically by aid of the integral calculus, we have

$$\triangle\,\varphi_1 = \int_{\tau_0}^{\tau_1} \frac{d\,\tau}{\tau}$$

$$= \log_e \frac{\tau_1}{\tau_0} \quad . \quad . \quad . \quad . \quad . \quad (3)$$



Also, since the temperature during boiling is constant and equal to τ_1, we have—

$$\triangle\,\varphi_2 = \Sigma \frac{\delta\,Q}{\tau_1} = \frac{1}{\tau_1}\,\Sigma\,\delta\,Q$$

$$= \frac{1}{\tau_1}\text{(sum of heat increments during boiling)}$$

$$= \frac{L_1}{\tau_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$\therefore$ Increase in entropy of steam over that of water at freezing point

$$= \triangle q_1 + \triangle q_2 = \log_e \frac{\tau_1}{\tau_0} + \frac{L_1}{\tau_1} \quad . \quad . \quad (5)$$

But if we agree to call the entropy of water at freezing point zero, the increase of entropy of water at τ_1 becomes equal to the entropy of water at τ_1, and the increase of entropy of steam becomes equal to the entropy of steam at τ_1, so that we have—

$$\text{Entropy of water at } \tau_1 \quad q_{w\tau_1} = \log_e \frac{\tau_1}{\tau_0} \quad . \quad . \quad . \quad . \quad (6)$$

$$\text{,, \quad steam at } \tau_1 \quad q_{s\tau_1} = \log_e \frac{\tau_1}{\tau_0} + \frac{L_1}{\tau_1} \quad . \quad . \quad . \quad (7)$$

Corresponding to these, we can plot points A and B on a diagram.

Similarly, if we took any other temperature τ_2, we should obtain corresponding points C and D, H C being equal to $\log_e \frac{\tau_2}{\tau_0}$ and C D equal to $\frac{L_2}{\tau_2}$.

In this way we can obtain any number of points corresponding to C and D, and on joining them we shall get two curves O C A and B D M, called the **Water and Steam Boundary Curves.**

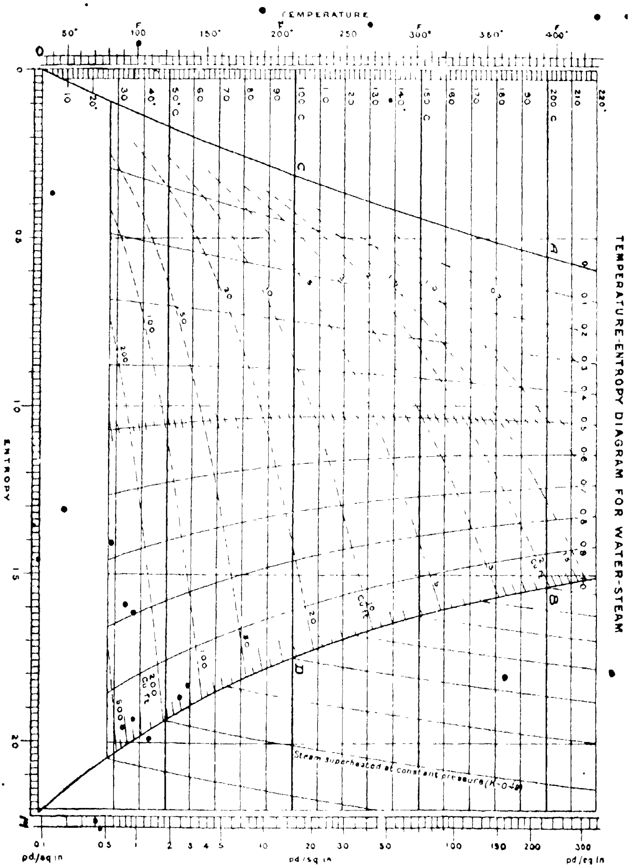
A point between the two curves represents steam in a "wet" condition, a point on the curve B D M represents steam in a dry saturated condition, while a point outside the curve B D M represents steam in a superheated condition. The wet and saturated conditions we will consider later. An entropy diagram to scale for steam is given below; this is reproduced from a diagram issued by H M Stationery Office which students are recommended to obtain.

Units of Entropy: the Rank. We will now consider the units in which entropy is to be measured.

It will be noted that the unit is of the order $\frac{\text{Quantity of Heat}}{\text{Temperature}}$, and since the heat units on the Fahrenheit and Centigrade scales bear the same ratio to each other as the absolute temperatures on these scales, we see that the entropy will have the same numerical values on the two scales. There is no universally agreed name for units of entropy, but Professor Perry has sug-

gested the name **Rank**, in commemoration of the great contributions of the late Professor Rankine to thermodynamics, and we will adopt that term

It will, of course, be understood that the properties of steam



vary slightly according to various authorities, and that the tabulated values of entropy of water and steam at various temperatures will vary correspondingly. Some authorities, for instance, have calculated the entropy of water based upon the slight variation in the specific heat of water found by Callendar and Barnes.

Numerical Example. Take, for instance, dry saturated steam at 200 lbs per square inch pressure absolute. From the tables on pp. 88, 89 of Vol. I. we have on the Fahrenheit scale $t = 381 \cdot 7^\circ$ F. and L = 843·8 B.Th.U.

$$\therefore \qquad \tau = 381 \cdot 7 + 459 \cdot 1 = 840 \cdot 8.$$

$$\therefore \qquad \phi = \log_e \frac{840 \cdot 8}{491 \cdot 1} + \frac{843 \cdot 8}{840 \cdot 8}$$

$$= 2 \cdot 3 \,(\log_{10} 840 \cdot 8 - \log_{10} 491 \cdot 1) + \frac{843 \cdot 8}{840 \cdot 8}$$

$$= \cdot 537 + 1 \cdot 003$$

$$= 1 \cdot 540 \text{ Ranks.}$$

On the Centigrade scale we shall have $t = 194 \cdot 3^\circ$ C., $\tau = 194 \cdot 3 + 273 = 467 \cdot 3^\circ$ C., and L = 468·8 C.H.U.

$$\therefore \qquad \phi = \log_e \frac{467 \cdot 3}{273} + \frac{468 \cdot 8}{467 \cdot 3}$$

$$= \cdot 537 + 1 \cdot 003$$

$$= 1 \cdot 540 \text{ Ranks.}$$

This example illustrates our statement that the numerical value of the entropy measure above the freezing point of water is the same on the Fahrenheit and the Centigrade scales.

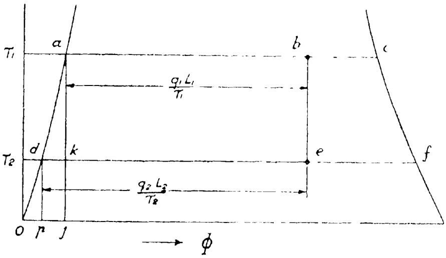
Dryness after Adiabatic Expansion. Suppose that we start with steam at temperature τ_1 and dryness coefficient q_1 and expand it adiabatically until the temperature is τ_2: we wish to ascertain the dryness of the steam after expansion.

We have seen already in Volume I. that if steam has a dryness coefficient or fraction equal to q_1, then in 1 lb. of wet steam there are q_1 lbs. of dry steam and $(1 - q_1)$ lbs. of wet steam; we may, therefore, assume that the latent heat of the wet steam is q_1 L$_1$ per pound. If, therefore, we take a point b on the entropy line $a\,c$ such that $\frac{a\,b}{a\,c} = q$; *i.e.*, $a\,b = \frac{q_1 \mathrm{L}_1}{\tau_1}$, b will represent

the entropy of 1 lb of steam at temperature τ_1 and dryness coefficient q_1.

If we now expand the steam adiabatically to temperature τ_2, the corresponding change in entropy diagram will be represented by the vertical line $b\,e$, and the dryness q_2 at this temperature will be given by the ratio $\frac{d\,e}{d\,f}$.

For purposes of practical calculation, we may either proceed as graphically above, using the chart reproduced upon p. 23, upon which lines will be noted corresponding to values of q from 0·1 to 0·9, or else employ the formula which we will now derive from a consideration of the temperature diagram.



$$d\,e = d\,k + k\,e = d\,k + a\,b = d\,k + \frac{q_1\,L_1}{\tau_1}$$

Now $d\,k = o\,j - o\,p = \log_e \frac{\tau_1}{\tau_0} - \log_e \frac{\tau_2}{\tau_0}$

$$= \log_e \tau_1 - \log_e \tau_0 - (\log_e \tau_2 - \log_e \tau_0)$$

$$= \log_e \tau_1 - \log_e \tau_2 = \log_e \frac{\tau_1}{\tau_2}$$

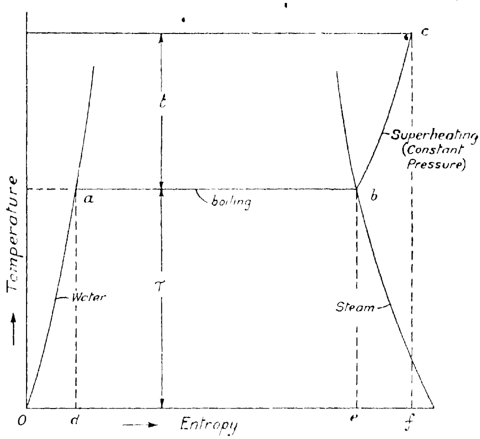
$\therefore \quad d\,e = \log_e \frac{\tau_1}{\tau_2} + \frac{q_1\,L_1}{\tau_1},$

but $d\,e = \frac{q_2\,L_2}{\tau_2}$.

We, therefore, obtain the following formula —

$$\frac{q_2\,L_2}{\tau_2} = \log_e \frac{\tau_1}{\tau_0} + \frac{q_1\,L_1}{\tau_1} \quad . \quad . \quad . \qquad (8)$$

Entropy Diagram for Superheated Steam. Now, suppose we have steam that is generated at temperature τ and is then superheated by a temperature t, the pressure remaining constant during the process of superheating.



Suppose that C_p, the specific heat of superheated steam, is constant* during the period of superheating. Then the entropy will increase in this period in manner indicated by the curve bc, and the increase of entropy ef will be given by

$$ef = \int_{\tau}^{\tau+t} C_p \frac{d\tau}{\tau} = C_p \log_e\left(\frac{\tau+t}{\tau}\right) \ldots$$

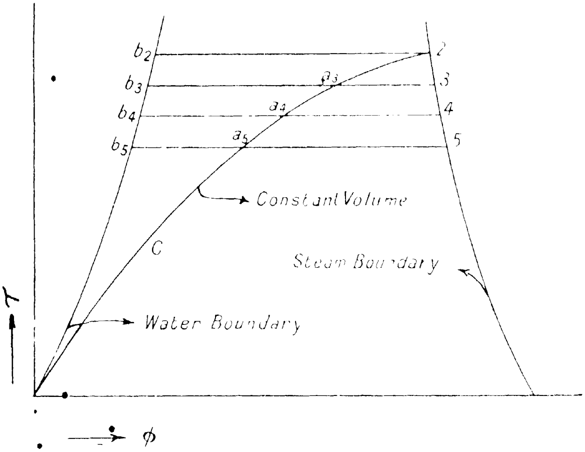
In the diagram reproduced on p. 23 there will be noticed a number of curves corresponding to bc on the right-hand side of the Steam Boundary Curve.

Constant Volume Lines on the Entropy Diagram for Steam.—On the diagram reproduced upon p. 23 will be found a number of curved lines corresponding to 2 C on the above diagram, the volumes in cubic feet per pound of steam being written against them.

* See Table on p. 57 of Vol. I for the variation of this quantity obtained from Mollier's researches.

To draw these lines approximately, we may proceed as follows —Take, for instance, the curve corresponding to a volume of 2 cubic feet per pound From the steam tables ascertain the absolute temperature corresponding to this specific volume, and draw the corresponding line $b_2 2$ upon the $\tau\phi$ diagram.

Draw similar lines $b_3 3$, $b_4 4$, $b_5 5$, etc., upon the diagram, and make $b_3 a_3 = \frac{2}{3} b_3 3$, $b_4 a_4 = \frac{1}{2} b_4 4$, $b_5 a_5 = \frac{2}{5} b_5 5$, and so on Then by joining up the points 2, a_3, a_4, etc., we obtain the constant volume line



Numerical Examples. (1) *A pound of water at 150° C. is converted into dry saturated steam; it is then expanded so that it always maintains the dry saturated condition until the temperature is 100° C. What is the increase in entropy during the expansion?*

If we refer to an indicator diagram we shall see that during expansion under the above condition the entropy diagram will follow the steam boundary line

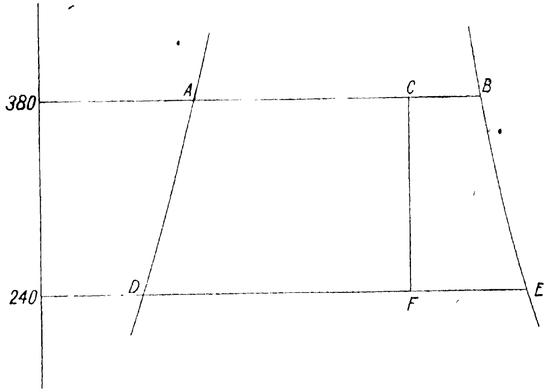
∴ Increase of entropy per pound = entropy at

150° C. − entropy at 100° C. = 1·74 − 1·62 = ·12 Rank

(2) *Wet steam at 380° F having a dryness coefficient equal to ·75 is expanded adiabatically until the temperature is 210° F What will be the dryness coefficient at the end of the expansion?*

We make a tracing of the relevant portion A B E D of the temperature-entropy chart, and make $AC = \cdot 75\, AB$, and then draw C F vertically to meet D E in F.

Then dryness coefficient after expansion $= \dfrac{DF}{DE} = \cdot 695.$



Clapeyron's Equation for a Relation between Volume and Pressure of Dry Saturated Steam.—Suppose that we have an imaginary steam engine working on the Carnot cycle between temperature limits which differ by a very small amount $\delta\tau$, the mean temperature being τ. Then if a unit mass of material is used, the indicator diagram will be as shown, V being the volume of a unit mass of saturated steam and w that of water.

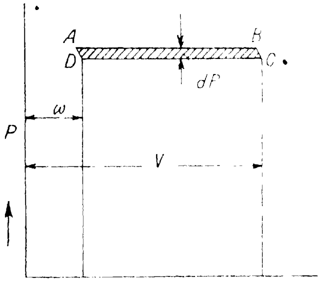
A B represents the isothermal, during which water changes into steam; B C the adiabatic, during which the temperature drops slightly and the pressure changes by an amount δP; C D the isothermal, during which the steam changes back to water, and D A the adiabatic compression, by means of which the water is brought back to its original condition.

Now, consider the corresponding $\tau\phi$ diagram; this will clearly be a horizontal strip as shown, and the shaded area in the two diagrams each represents the work done in the cycle, so that these areas must be equal.

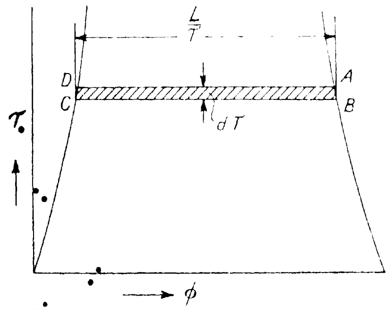
We, therefore, have—

$$(V - w)\, d\,P = \frac{L}{\tau}\, d\,\tau,$$

$$\text{or } (V - w) = \frac{L}{\tau}\,\frac{d\,\tau}{d\,P}.$$



Indicator Diagram



Entropy Diagram

This is known as *Clapeyron's equation*, and it enables us to calculate the volume of steam at a given temperature when we know the latent heat, and the rise of pressure for one degree rise in temperature at that temperature.

Lecture II.—Questions.

1. A pound of air of volume 3 cubic feet; $p = 15{,}950$ lbs. per square foot; $\tau = 900°$ F. expands at constant temperature until its volume is 12 cubic feet. Find the new pressure and the loss or gain in entropy. *Ans.* Pressure = 3,988 lbs./ft.; entropy gain = ·095.

2. Steam is at 90 lbs. per square inch pressure ($t = 320°$ F., $h = 888·4$), and the change of pressure per °F. = 1·28 lbs. per square inch. Find the volume of dry steam per lb., taking $w = ·016$. *Ans.* 4·80 cubic feet approximately.

3. Calculate the entropy of 1 lb. of superheated steam generated at a temperature of 341° F. and superheated 120° F. *Ans.* 1·65.

4. A heat engine receives 1,000 B.Th.U. at 300° F. and 500 B.Th.U. uniformly as temperature falls to 200° F. All the heat is rejected at 125° F. Find the maximum possible efficiency of the engine. *Ans.* 21·4 per cent.

5. Find, by means of the $\tau\phi$ chart, the efficiency of a steam engine which is jacketed so that the steam remains saturated throughout the expansion.

6. Dry steam at 105 lbs. per square inch (abs.) expands to 15 lbs. per square inch (abs.), and is still dry. How much heat must have been added per lb.?

7. Prove that when a gas at temperature T_2 and volume v_2 changes to temperature T_1 and volume v_1, the change in entropy is

$$K_v \log_e \frac{T_2}{T_1} + (K_p - K_v) \log_e \frac{v_2}{v_1}$$

8. Show that the volume of 1 lb. of saturated steam may be obtained from the formula $u = v + \frac{J\,L}{T}\frac{dT}{dP}$, where

- u = the volume of the steam in cubic feet per pound.
- v = volume of 1 lb. of water in cubic feet.
- J = Joule's equivalent.
- L = latent heat of steam.
- T = absolute temperature of steam.

Use the following table to obtain $\frac{dP}{dT}$, and find the volume of 1 lb. of steam at 160 lbs. absolute pressure per square inch.

Absolute pressure pounds per square inch.	159	160	161
Temperature Fahrenheit degrees, .	363·1	363·6	364·1
Temperature Centigrade degrees,	183·9	184·2	184·5

The latent heat of 1 lb. of steam at 160 lbs. pressure is 858·8 B.Th.U., or 477 C.H.U.

Lecture II.—A.M.Inst.C.E. Questions.

1. Describe the ideal Carnot cycle by means of a $p\,v$ diagram, and by means of a temperature-entropy diagram. A perfect engine working on the Carnot cycle receives 5,000 B.Th.U. per minute at 2,000° F.; the heat is rejected at 500° F. Find the horse-power and the efficiency of the engine.

2. Sketch a $\tau\,\phi$ chart for steam and show by its means that the loss due to reducing the admission pressure by throttling is considerably less than that caused by an equal increase in back pressure.

3. Prove the equation $V = w + J\dfrac{L\,d\tau}{\tau\,dp}$, by which the volume per pound of dry saturated steam may be deduced from a knowledge of the latent heat—temperature and the pressure—temperature curves and the density of water. Calculate the volume per pound of dry saturated steam at 200 lbs. per square inch absolute from the following data:—

Pressure (lbs. per square inch absolute),	195	200	205
Temperature, ° F., .	379.4	381.6	383.7

The latent heat at pressure 200 is 850.3 B.Th.U.

4. Show that the work done in compressing adiabatically 1 lb. of a perfect gas is the same so long as the ratio of the initial to the final pressure is constant.

5. Show by the help of a temperature-entropy chart how superheating effects economy in a steam engine, and the point beyond which superheating might be expected to cause practical difficulties greater than any increase in the economy would compensate for.

6. Given the following data, construct the temperature-entropy diagram for 1 lb. of dry steam at 100 lbs. initial absolute pressure, generated from feed-water at 142° F., expanded adiabatically to 3 lbs. per square inch absolute pressure, and exhausted at that pressure.

Pressure Absolute	Temperature	Entropy of 1 lb. of Water	Entropy of 1 lb. of Steam
lbs. per sq. inch	Degrees F.		
3	142	0.201	1.887
100	328	0.476	1.596

Also find, in the above example, (*a*) the work done per pound of steam, and (*b*) the dryness of the steam after expansion.

LECTURE III.

ENTROPY—*Continued.*

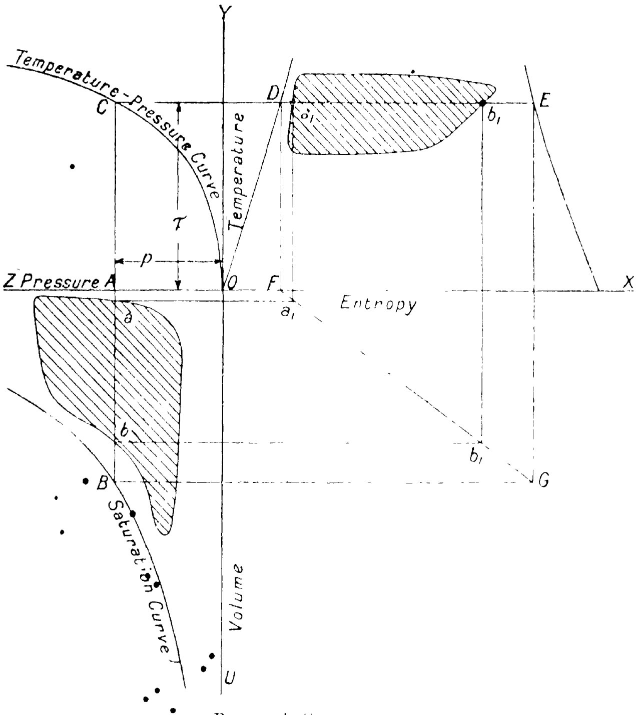
CONTENTS.—Conversion of Indicator Diagram for a Steam Engine to a Temperature-Entropy Diagram—Boulvin's Construction—Mollier's Diagram for Steam—Application of Mollier's Diagram to Throttling Steam—Application of Mollier's Diagram to Steam Turbines—Temperature-Entropy Diagrams for other Vapours—Increase of Entropy in Irreversible Processes—Questions.

Conversion of Indicator Diagram for a Steam Engine to a Temperature-Entropy Diagram. It is interesting to find the temperature-entropy diagram corresponding to a given indicator diagram; at one time it was thought that this would be of great assistance in giving information as to the behaviour of the steam in the cylinder, and that it would become the standard practice to affect this conversion. As we shall see later, however, this can only be done from a knowledge of the saturation curve for the expansion of the steam, and on the assumption that the dryness of the steam is given by the ratio of the actual volume to the saturation volume; except in the case of research work, therefore the drawing of entropy diagrams is not very often carried out in practice. It is clearly difficult to argue with much accuracy upon the action of the steam in a cylinder from the form which the entropy diagram takes when obtained in this necessarily indirect manner. We have also to remember that the indicator diagrams are themselves liable to considerable error. If we had any means of drawing an entropy diagram for an engine by direct means, the value of the diagram would be much increased.

The student should, however, make himself familiar with the procedure to be adopted in the conversion of indicator to temperature diagrams; in addition to being very interesting, the study has the merit of giving considerable insight into the subject and of focussing the student's ideas.

Boulvin's Construction.—In this construction we take two lines Y O U and Z O X at right angles and in one compartment, conveniently Z O U, we draw the indicator card and the saturation curve with volume axis O U and pressure axis O Z. The curve

is obtained by plotting from steam tables the volume of saturated steam for various pressures, the weight of steam passing into the cylinder per stroke being known. Alongside this curve we draw in the compartment Z O Y a curve of pressures plotted against temperatures, O Z being the pressure axis and O Y the tempera-



BOULVIN'S CONSTRUCTION.

ture axis; this curve is also plotted from the steam tables, the pressure scale being the same as for the indicator diagram. In the compartment Y O X we draw the entropy boundaries for water and steam, the temperature scale being the same

as for the pressure-temperature curve Now draw a vertical line A B across the indicator card and project A vertically to intersect the pressure-temperature curve in the point C, and then project this point horizontally to intersect the water and steam entropy boundaries in D and E. A vertical should next be drawn through D to intersect O X in F, and a vertical should be drawn through E to intersect in G a horizontal line drawn through B, and join F G Now the line A B intersects the indicator card in a and b, and we may assume approximately that

$$\frac{A\,b}{A\,B} = \text{Dryness coefficient of steam at } b.$$

$$\frac{A\,a}{A\,B} = \text{Dryness coefficient of steam at } a.$$

Now project b and a horizontally to meet F G in a_1 and b_1, and then project a_1 and b_1 vertically to intersect D E in a_1 and b_1.

Then a_1 and b_1 are the points on the temperature entropy diagram corresponding to a and b on the indicator diagram. By drawing a number of lines on the indicator diagram parallel to A B, we obtain corresponding points on the temperature-entropy diagram, and by joining up these points we are able to obtain the closed figure illustrating the variation in entropy throughout the cycle

To prove the construction, we note that

$$\frac{D\,a_1}{D\,E} = \frac{F\,a_1}{F\,G} = \frac{A\,a}{A\,B} = \text{Dryness coefficient of steam at } a = x_a.$$

But we have seen that if we have steam with dryness coefficient x_a and a_1 is the corresponding point on the isothermal line D E—

$$\frac{D\,a_1}{D\,E} = x_a.$$

∴ a_1 is the point on the entropy diagram corresponding to a on the indicator diagram

Similarly, we may prove that b_1 is the point on the entropy diagram corresponding to b on the indicator diagram

Mollier's Diagram for Steam. By plotting total heat against entropy we obtain a diagram called a Mollier diagram.*

* A useful form of this diagram to enlarged scale, upon which the diagram on p. 36 is based, is published by Messrs. Oliver & Boyd, of Edinburgh. Price 6d. net. Students are recommended to obtain the enlarged diagram and employ it in their calculations.

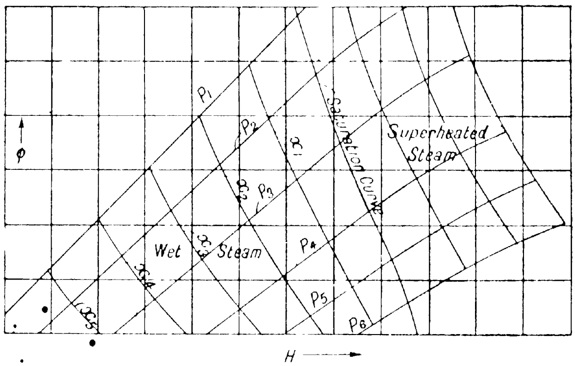
A number of lines of constant pressure are drawn across the diagram, five of which, P_1 ... P_5, we have indicated on the figure, and a saturation curve is also drawn. Then points on the left-hand side of the saturation curve represent wet steam, and those on the right-hand side represent superheated steam.

We have seen already that the total heat of steam at a given pressure and of dryness coefficient x may be taken as given by

$$H = s + x L \qquad . \qquad . \qquad (1)$$

where

H = total heat,
s = sensible heat,
L = latent heat.



and

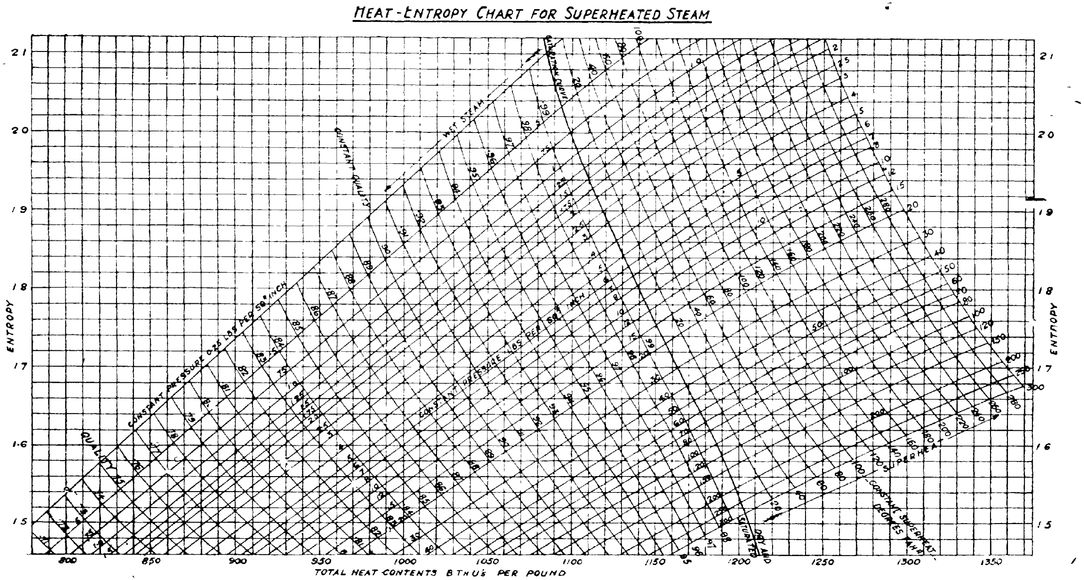
$$\phi = \log_e (\tau / \tau_0) + \frac{x L}{\tau} \qquad (2)$$

$$= \log_e (\tau / \tau_0) + \frac{(H - s)}{\tau} \qquad (3)$$

$\therefore$ since τ, s, and τ_0 are fixed quantities, it will be seen that the diagram of H plotted against ϕ when the pressure is constant will be straight lines.

We thus see that the constant pressure lines on the left-hand side of the saturation line will be straight lines. We can plot across these lines curves representing given values of the dryness coefficient.

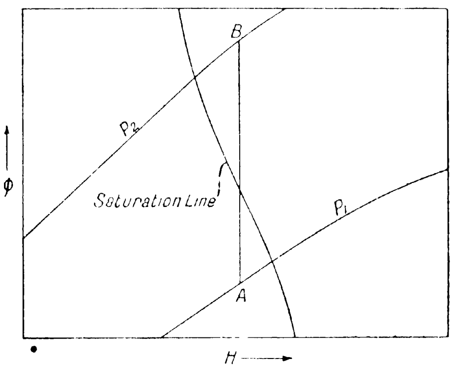
On the other side of the saturation curve the constant pressure



MOLLIER'S DIAGRAM FOR STEAM (Fahrenheit Scale).

lines become somewhat curved, as shown, and corresponding to the dryness coefficient lines on the wet-steam portion of the diagram we can draw lines of constant superheat temperature on the right-hand portion of the diagram

Application of Mollier's Diagram to Throttling Steam.—The process of throttling steam involves a diminution of pressure without loss of total heat, we can thus from a Mollier diagram ascertain the condition of steam initially at a given pressure and of known dryness, when throttled to a new pressure, by drawing a vertical line on the diagram. Suppose that the initial pressure is P_1, and that the point on the P_1 line corresponding to the known dryness fraction is A, to ascertain the condition of the



Total Heat.

steam after throttling to a pressure P_2, we draw a vertical line A B until we intersect the P_2 line at the point B. From the diagram we can then find the dryness at the new pressure; if the vertical line crosses the saturation curve we see that the steam in its new condition will be superheated

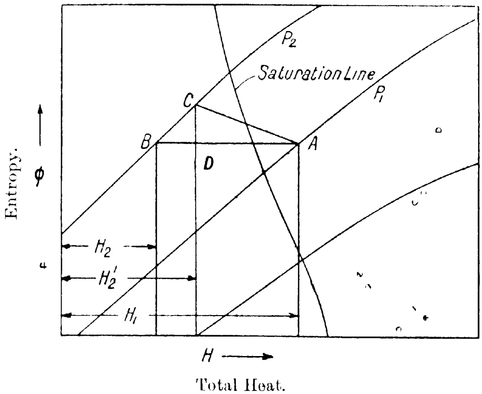
Throttling Calorimeter.—Calculations upon the throttling calorimeter can be made by a similar use of constant heat lines upon the Mollier diagram In the use of this instrument we measure the absolute pressure of steam, P_1 and P_2, before and after throttling, and also the temperature, t_2, after throttling. By means of the steam tables we can ascertain the saturation tem-

perature corresponding to the pressure P_2, and thus by subtraction find the degree of superheat. We can thus find the point corresponding to β on the Mollier diagram, and by dropping a vertical to meet the P_1 line we read off the initial dryness coefficient of the steam.

Numerical Example. Find the dryness coefficient of steam from the following readings taken with a throttling calorimeter:—Pressure in main steam pipe, 150 lbs. per square inch absolute; pressure after wire-drawing, 17·2 lbs. per square inch absolute; temperature after wire-drawing, 240° F.

The saturation pressure corresponding to 17·2 lbs. per square inch absolute = 220° F., so that we have 20° F. of superheat. From the diagram we then find the point corresponding to 17·2 lbs. per square inch pressure, and a superheat of 20° F., and drop vertically until we reach the pressure curve of 150 lbs. per square inch. We then read off the dryness as approximately ·97. It is useless to attempt great accuracy in calculation upon dryness of steam, because the necessary errors made in the observations are considerable.

Application of Mollier's Diagram to Steam Turbines. The effect of friction upon the flow of steam through a steam turbine can be studied by the aid of Mollier's diagram.



Total Heat.

In the ordinary calculations we assume that the expansion of steam in passing through a nozzle is adiabatic—*i.e.*, that the entropy is constant.

Let A represent the point corresponding to the initial pressure P_1 of the steam, and H_1 the initial total heat of the steam. Then if the process of expansion to pressure P_2 were truly adiabatic, the entropy would remain constant throughout the operation and the process would be represented by the straight line A B, the total heat at the end of the operation being H_2; the quantity $(H_1 - H_2)$ is called the heat drop. Owing, however, principally to friction, the actual expansion process follows a line A C, the entropy being increased and the effective heat drop, from which the steam velocity is calculated, becomes $(H_2{}^1 - H_1)$, D B representing the loss of heat due to friction.

If we know from experimental results for a given case the proportion of the friction heat loss to the total heat drop available, *i.e.*, the ratio $\frac{DB}{AB}$, we can find the point D and then obtain C by drawing D C until we meet the P_2 line.

It will be noted that the vertical line D C represents a throttling action during which the total heat is constant, so that we may regard the process of expansion with friction as equivalent to a frictionless adiabatic expansion down to the pressure corresponding to the point D, and then a throttling down to the pressure P_2.

Temperature-Entropy Diagrams for Other Vapours. From a knowledge of the properties of other saturated vapours we are able to draw their temperature-entropy diagrams. The following tables deal with the gases ammonia, carbon dioxide, and sulphur dioxide, and are of interest in problems concerning refrigeration, with which we deal in Chapter VI.

Properties of Saturated Vapours. The following data are derived from tables given in Zeuner's "Technical Thermodynamics".

P = pressure in pounds per square inch, absolute ;
t = temperature Fahrenheit ;
V = volume of 1 lb., cubic feet ;
S = sensible heat in the liquid above 32° F., B.Th.U. ;
H = total heat above 32° F., B.Th.U. ;
L = heat of vaporisation = H − S, B.Th.U. ;
φ_l = entropy of the liquid at the boiling point, above 32° F. ;
φ_v = entropy of vaporisation = $\frac{L}{T}$;
φ_t = total entropy of the dry vapour = $\varphi_l + \varphi_v$.

PROPERTIES OF DRY SATURATED AMMONIA.

t	P	V	s	L	H	ϕ_l	ϕ_v	ϕ_t
−22	16·912	15·956	−47·88	593·8	545·92	−0·1029	1·3576	1·2547
−13	21·456	12·767	−40·84	590·4	549·56	−0·0870	1·3225	1·2355
− 4	27·030	10·306	−33·43	586·4	552·97	−0·0706	1·2879	1·2173
5	33·667	8·390	−25·63	582·1	556·47	−0·0536	1·2536	1·2000
14	41·571	6·888	−17·46	577·4	559·94	−0·0362	1·2198	1·1836
23	50·904	5·699	− 8·93	572·4	563·47	−0·0183	1·1864	1·1681
32	61·836	4·709	0	557·5	557·50	0	1·1344	1·1344
41	74·539	3·933	10·12	548·5	558·62	0·0203	1·0943	1·1146
50	89·195	3·300	20·30	537·5	557·80	0·0405	1·0544	1·0949
59	105·974	2·787	30·60	526·7	557·30	0·0660	1·0132	1·0792
68	125·056	2·371	41·04	515·9	556·94	0·0804	0·9741	1·0545
77	146·612	2·026	51·48	503·8	555·28	0·1001	0·9359	1·0360
86	170·807	1·738	62·10	490·5	552·60	0·1196	0·8984	1·0180
95	197·796	1·493	72·72	474·8	547·52	0·1391	0·8636	1·0027
104	227·797	1·296	83·48	461·9	545·38	0·1583	0·8252	0·9837

PROPERTIES OF DRY SATURATED CARBON DIOXIDE.

t	P	V	Specific Volume of the Liquid	S	L	H	φ_l	φ_v	φ_t
−22	213	0·432	0·0155	−24·80	126·72	101·92	−0·053	0·290	0·236
−13	249	0·367	0·0157	−21·06	123·25	102·19	−0·045	0·276	0·231
− 4	289	0·312	0·0160	−17·19	119·43	102·24	−0·036	0·262	0·226
5	334	0·267	0·0163	−13·18	115·25	102·07	−0·028	0·248	0·221
14	385	0·228	0·0167	− 9·00	110·65	101·65	−0·019	0·234	0·215
23	441	0·195	0·0171	− 4·63	105·53	100·90	−0·010	0·219	0·209
32	504	0·167	0·0176	0	99·81	99·81	0	0·203	0·203
41	573	0·143	0·0181	4·93	93·35	98·28	0·010	0·187	0·197
50	650	0·120	0·0187	10·28	85·93	96·21	0·021	0·169	0·189
59	734	0·101	0·0197	16·22	77·20	93·42	0·032	0·149	0·181
68	826	0·083	0·0210	23·08	66·47	89·55	0·045	0·126	0·171
77	930	0·067	0·0227	31·63	52·16	83·79	0·061	0·097	0·159
86	1,040	0·048	0·0268	45·45	27·00	72·45	0·087	0·050	0·136
87·8	1,062	0·042	0·0298	51·61	15·12	66·73	0·098	0·028	0·126
88·43	1,071	0·035	0·0346	59·24	0	59·24	0·112	0	0·112

PROPERTIES OF DRY SATURATED SULPHUR DIOXIDE.

t	P	V	S	L	H	φ_l	φ_r	φ_t
−22	5.564	12.720	−16.675	172.597	155.922	−0.0359	0.3946	0.3589
−13	7.228	10.074	−13.957	171.950	157.993	−0.0298	0.3842	0.3554
−4	9.272	8.051	−11.216	171.000	159.784	−0.0237	0.3755	0.3518
5	11.756	6.486	−8.449	169.745	161.296	−0.0177	0.3655	0.3478
14	14.745	5.265	−5.657	168.187	162.530	−0.0117	0.3553	0.3436
23	18.311	4.304	−2.842	166.325	163.483	−0.0058	0.3448	0.3390
32	22.530	3.540	0	164.160	164.160	0	0.3341	0.3341
41	27.483	2.930	2.866	161.690	164.556	0.0058	0.3231	0.3289
50	33.253	2.436	5.758	158.917	164.675	0.0115	0.3120	0.3235
59	39.931	2.037	8.674	155.840	164.514	0.0172	0.3006	0.3178
68	47.611	1.711	11.615	152.460	164.075	0.0228	0.2891	0.3119
77	56.386	1.445	14.582	148.775	163.357	0.0284	0.2773	0.3057
86	66.359	1.220	17.572	144.787	162.359	0.0339	0.2655	0.2994
95	77.630	1.036	20.588	140.495	161.083	0.0394	0.2534	0.2928
104	90.300	0.884	23.629	135.900	159.529	0.0448	0.2412	0.2860

PROPERTIES OF DRY SATURATED CARBON DIOXIDE.

t	P	V	Specific Volume of the Liquid	S	L	H	ϕl	ϕv	ϕt
−22	213	0·432	0·0155	−24·80	126·72	101·92	−0·053	0·290	0·236
−13	249	0·367	0·0157	−21·06	123·25	102·19	−0·045	0·276	0·231
−4	289	0·312	0·0160	−17·19	119·43	102·24	−0·036	0·262	0·226
5	334	0·267	0·0163	−13·18	115·25	102·07	−0·028	0·248	0·221
14	385	0·228	0·0167	−9·00	110·65	101·65	−0·019	0·234	0·215
23	441	0·195	0·0171	−4·63	105·53	100·90	−0·010	0·219	0·209
32	504	0·167	0·0176	0	99·81	99·81	0	0·203	0·203
41	573	0·143	0·0181	4·93	93·35	98·28	0·010	0·187	0·197
50	650	0·120	0·0187	10·28	85·93	96·21	0·021	0·169	0·189
59	734	0·101	0·0197	16·22	77·20	93·42	0·032	0·149	0·181
68	826	0·083	0·0210	23·08	66·47	89·55	0·045	0·126	0·171
77	930	0·067	0·0227	31·63	52·16	83·79	0·061	0·097	0·159
86	1,040	0·048	0·0268	45·45	27·00	72·45	0·087	0·050	0·136
87·8	1,062	0·042	0·0298	51·61	15·12	66·73	0·098	0·028	0·126
88·43	1,071	0·035	0·0346	59·24	0	59·24	0·112	0	0·112

Lecture III.—Questions.

1. Draw $\tau\phi$ diagrams for carbonic acid and ammonia.

2. Dry steam at 300 lbs. per square inch absolute is generated in a boiler and then passed through reducing valve and throttled to 250 lbs. per square inch without loss of heat. Find the dryness at the lower pressure side of the reducing valve, or, if the steam be superheated, find the number of degrees of superheat. *Ans.* 10·2° F. superheat.

3. Explain how superheating increases the efficiency of a steam engine.

4. Describe any form of throttling calorimeter, and point out the difficulty of getting good results with steam calorimeters generally.

5. In the use of a throttling calorimeter the gauge pressure was 80 lbs. per square inch, atmospheric pressure 29·5 inches mercury; manometer reading 2 inches mercury; thermometer reads 220° F. Find the wetness. Steam tables may be used. *Ans.* Wetness = 3·8 per cent.

6. Steam is wire-drawn from 300 lbs. pressure ($t_1 = 417°$) to 200 lbs. pressure ($t_2 = 381°$). If the steam contains 10 per cent. of water, find the amount of drying produced.

7. Show how to combine in one diagram the indicator diagrams from a compound steam engine, and afterwards transfer them to a temperature-entropy chart.

8. In a combined separating and wiredrawing calorimeter the following observations were taken:—Total quantity of steam passed through the diaphragm, 52 lbs.; water drained from the separator, 2·7 lbs.; steam pressure before wire-drawing, 118 lbs. square inch absolute (temperature 340° F.) (171·1° C.), latent heat 878·3 B.T.U., 488 C.H.U. Temperature of steam on leaving, 232·6° F. (111·4° C.). Steam pressure on leaving, atmospheric. Find the wetness fraction of the steam on entry. You may take the specific heat of superheated steam as 0·48.

9. Steam of wetness fraction w_1 is expanded adiabatically from a pressure p_1 to a pressure p. The expansion curve is $p\,v^n =$ constant, where $n = 1{\cdot}135 - 0{\cdot}1\,w$ and the specific volume of steam is given by $P\,V^{\frac{16}{15}} =$ constant. Find the wetness of the steam after expansion from p_1 to p in terms of p_1, p, and w_1. Hence find the wetness of 5 per cent. wet steam after adiabatic expansion from 150 to 20 lbs. per square inch absolute.

10. Show, by sketches of the "total heat-entropy," or "Mollier," diagram, how the following required data may be determined:—(*a*) The dryness or superheat of steam which expands through a throttling valve without loss of total heat; (*b*) the work done by 1 lb. of steam expanding adiabatically from a given pressure and with a given amount of superheat or wetness to a lower pressure; (*c*) the dryness of steam after expansion in a nozzle when the efficiency of the nozzle is known; (*d*) the original condition of steam which has passed through a throttling calorimeter, when the pressure and superheat of the steam are known at the end of the expansion.

LECTURE III.—A.M.INST.C.E. QUESTIONS.

1. Find the dryness of the steam after cut-off at three-quarters of the stroke from the following particulars of an engine trial, assuming no leakage :—

Condensed steam per hour, in pounds, .	1,608
Revolutions per minute, .	120
Volume of cylinder, cubic feet,	3·6
Clearance per cent., . .	5
Pressure of steam in pounds per square inch at $\frac{3}{4}$ stroke,	41·8
Volume in cubic feet of 1 lb. of steam at 41·8 lbs. pressure, .	10·05
Pressure of steam in pounds per square inch at 0·84 of the return stroke and commencement of compression, .	17·2
Volume in cubic feet of 1 lb. of steam at 17·2 lbs. per square inch	23·14

2. Steam passes through a throttling calorimeter, where it is reduced in pressure from 120 lbs. per square inch (temperature 341° F.) to 15 lbs. per square inch. The temperature after expansion is 230° F. The temperature of steam at 15 lbs. per square inch is normally 213° F. Find the original dryness of the steam. The latent heat of 1 lb. of dry steam is approximately 1,114 – 0·7 t thermal units, where t is the temperature of the steam in degrees Fahrenheit. The specific heat of superheated steam may be taken as 0·5.

3. State the second law of thermodynamics, and show how the growth of entropy in irreversible cycles is in conformity with this law.

4. Draw a Mollier heat-chart with the data given below (only the 200° F. superheat line, the saturation line, and the lines representing the given pressures need be drawn). Why are the pressure lines straight in the saturated field and curved in the superheated field? Assuming the initial condition of the steam to be 200 lbs. per square inch absolute pressure and 200° F. superheat, draw a line representing the Rankine cycle and a line giving 70 per cent. efficiency ratio, in both cases terminating at 2 lbs. absolute pressure.

	Pressure	Total Heat	Entropy.
	lbs per Sq In abs	B.Th.U per Lb	
Superheat 200° F.,	200	1,308	1·663
	100	1,289	1·719
	25	1,256	1·833
	5	1,222	1·972
	1	1,206	2·053
Saturated,	200	1,198	1·546
	100	1,186	1·602
	25	1,160	1·714
	5	1,130	1·843
	1	1,115	1·918
0·8 dryness fraction,	200	1,030	1·346
	100	1,009	1·377
	25	970	1·444
	5	930	1·520
	1	911	1·570

5. The dryness fraction of steam is ascertained to be 0·96, and the pressure is 170 lbs. per square inch absolute Find the total heat of this steam given that the total heat of saturated steam at this pressure is 1,195·4 B.Th.U per pound, and that the water heat is 340·7 B.Th.U per pound. If this steam were expanded by throttling, show how to calculate the temperature when the steam is just saturated

6 Explain the statement that if the condition of a substance is changed along a reversible path the difference of entropy between the final and the initial stages is the summation of $\frac{dQ}{T}$. Show that the difference of entropy is greater than this if the path is irreversible

7 Show how the entropy diagram for a steam engine may be used to investigate the amount of moisture present at various points in the revolution.

LECTURE IV

STANDARD THERMODYNAMIC CYCLES.

CONTENTS.—Four Standard Thermodynamic Cycles—Constant Volume Cycle—Stirling's Cycle—Otto or Beau de Rochas Cycle—Efficiency in Constant Volume Cycle—Air Standard Efficiency—Effect of Variable Specific Heat of Gas—Numerical Value of Specific Heat—Effect of Variable Specific Heat upon Adiabatic Expansion—Wimperis' Formula for the Ideal Efficiency of the Constant Volume Cycle with Variable Specific Heats—Comparison of Ideal Efficiencies for Constant and Variable Specific Heats—Constant Pressure Cycle—Rankine-Clausius Cycle—Efficiency without Superheat from Consideration of the pV and $\tau\phi$ diagrams—Comparison with Carnot Efficiency—Rankine-Clausius Cycle with Superheated Steam—Questions.

THERE are a number of standard ideal thermodynamic cycles, with which the cycle of any given engine can be compared as a standard. In dealing with these cycles, we will in most cases designate them according to their thermodynamic characteristics, under which the heat is taken in and rejected, and also by the names of the investigators with whom they are usually associated.

The following standard cycles require investigation, the first of which we have already considered in detail:—

(1) Constant Temperature or Carnot Cycle.
(2) Constant Volume or Stirling and Otto Cycles.
(3) Constant Pressure or Ericsson and Joule Cycles.
(4) Rankine-Clausius Steam Engine Cycle.

CONSTANT VOLUME CYCLE.

In this cycle all the heat is received at constant volume and rejected at constant volume. There are two principal forms of this cycle, in one of which (Stirling) the expansion and compression are isothermal, and in the other (Otto) they are adiabatic.

(*a*) *Stirling Cycle.*—In Stirling's air engine, invented in 1827, we have the first example of a "reversible" engine, and a good attempt at the employment of the regenerator principle.

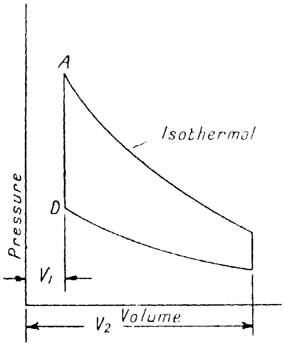
The cycle was as follows :

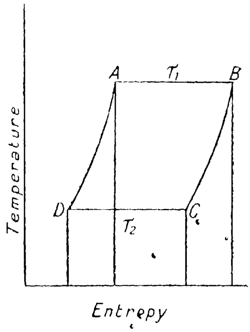
(1) Air at volume V_1, heated by passing through the regenerator, was allowed to expand isothermally to volume V_2, heat being taken in from a furnace, and the expansion causing a piston to be raised.

(2) The heated air was then passed back through the regenerator from the hot to the cold end, and gave up some of its heat with corresponding drop in pressure, the volume remaining constant.

(3) The air was then compressed isothermally to its original volume.

(4) The compressed cold air then passed back through the regenerator from the cold to the hot end, and re-absorbed heat previously given up.

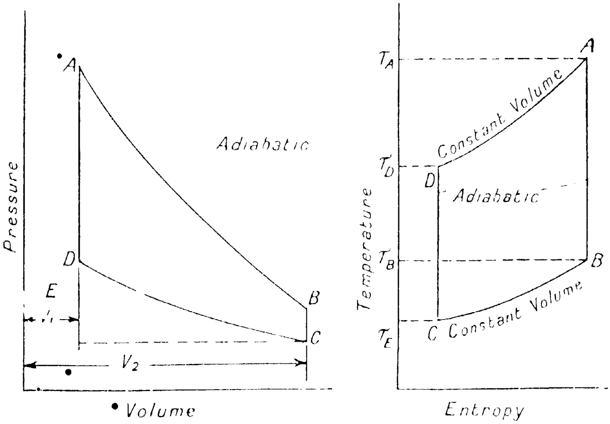




A Stirling engine of 50 I.H.P. was used in 1843 at Dundee, and gave a relatively high efficiency of ·3 measured on the I.H.P. But the mechanical efficiency was low, so that the over all efficiency was not so good. The objection to hot air engines from the practical point of view is that they have to be very bulky—that is to say, the power output of an engine of given size is small. Stirling attempted to overcome this objection by using high pressures that is, a relatively high weight of air per stroke, even for the lower pressure of the cycle but even if this be done the hot-air engine is not a practical proposition.

(*b*) *Otto or Beau de Rochas Cycle.* This cycle differs from the Stirling cycle in that the expansion and compression are adiabatic in place of isothermal, and is usually regarded as the ideal to which the Otto or Beau de Rochas explosion cycle approximates; the details of this cycle are given in Chapter IX. The above figure gives the pV and $\tau\phi$ diagrams for the theoretical cycle.

The charge is taken in at the point C and is compressed adiabatically to D; it is then exploded with immediate rise of pressure to A, and then expands adiabatically to B, where the exhaust opens and the pressure falls immediately.



In the four-stroke cycle the engine makes two idle strokes, represented by the dotted line, during which the products of combustion are expelled from the cylinder and a new charge is drawn in.

Efficiency in Constant Volume Cycle.

Heat supplied at constant volume from D to A

$$= C_v(\tau_A - \tau_D). \quad . \quad . \quad . \quad . \quad (1)$$

Heat rejected at constant volume from B to C

$$= C_v(\tau_B - \tau_C) \quad . \quad . \quad . \quad . \quad (2)$$

∴ Heat converted into work = heat supplied − heat rejected,

$$= C_v\{(\tau_A - \tau_D) - (\tau_B - \tau_C)\}$$

$$\therefore \quad \text{Efficiency} = \eta = \frac{\text{Heat converted into work}}{\text{Heat supplied}}$$

$$= \frac{C_v\{(\tau_A - \tau_D) - (\tau_B - \tau_C)\}}{C_v(\tau_A - \tau_D)}$$

$$= 1 - \frac{\tau_B - \tau_C}{\tau_A - \tau_D} \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

But by Equation (10), p. 11, since the expansion is adiabatic,

$$\frac{\tau_A}{\tau_B} = \frac{\tau_D}{\tau_C} = \left(\frac{V_2}{V_1}\right)^{\gamma-1} \quad . \quad . \quad . \quad . \quad (4)$$

$$\therefore \quad \frac{\tau_A}{\tau_D} = \frac{\tau_B}{\tau_C} \text{ or } \frac{\tau_A}{\tau_A - \tau_D} = \frac{\tau_B}{\tau_B - \tau_C},$$

$$\text{i.e.,} \quad \frac{\tau_B - \tau_C}{\tau_A - \tau_D} = \frac{\tau_B}{\tau_A} = \frac{\tau_D}{\tau_C} \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$\therefore \quad \eta = 1 - \frac{\tau_B}{\tau_A} = 1 - \frac{\tau_D}{\tau_C} \quad . \quad . \quad . \quad (6)$$

$$\text{Or} \quad \eta = 1 - \left(\frac{V_1}{V_2}\right) = 1 - \left(\frac{1}{r}\right)^{\gamma-1}, \quad . \quad (7)$$

where $r = \left(\frac{V_2}{V_1}\right)$ = ratio of expansion

This quantity $1 - \left(\frac{1}{r}\right)^{\gamma-1}$ is called the *air-standard efficiency*, γ being taken 1·40, and it is common to express the results of tests upon internal combustion engines with reference to this ideal efficiency in much the same way as with reciprocating steam engines. The Rankine-Clausius cycle (p. 6) is adopted as the standard

We thus have :—

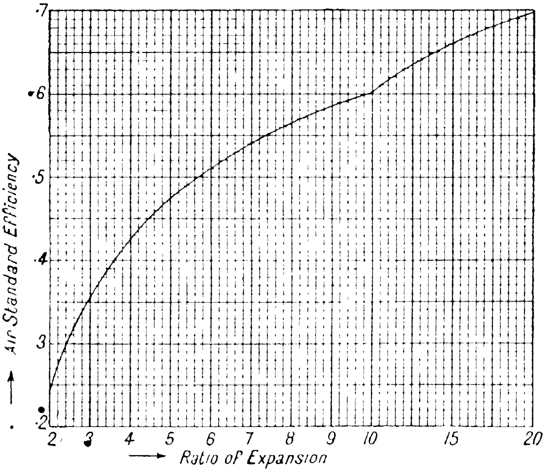
$$\text{Air standard efficiency} = \eta_a = 1 - \left(\frac{1}{r}\right)^{0 \cdot 4}.$$

This gives the following values :—

Ratio of Expansion	2	3	4	5	7	10	20	100
Air standard efficiency,	·242	·356	·426	·475	·541	·602	·698	·841

The above diagram enables the efficiencies for intermediate values of the expansion ratio readily to be ascertained.

It will be noted that the efficiency given by equation (6) is less than that for a perfect engine $\left(\text{which would be } 1 - \frac{T_C}{T_A}\right)$, because the heat is not all taken in at the same temperature and all rejected at the same temperature.



Numerical Example. A gas engine of 25 H.P. receives ·06 cubic foot of gas per cycle, the calorific value of which is 600 B.Th.U. per cubic foot. The clearance volume is 0·2 cubic foot, and the volume swept out by the piston each stroke is 1 cubic foot. Number of cycles per minute = 90. Find the efficiency of the engine and its efficiency ratio compared with the air standard.

In this case $V_1 = 0{\cdot}2$; $V_2 = 1{\cdot}2$. $\therefore r = 6$.

$\therefore$ Efficiency on air standard (from diagram) = ·52 approx.

Heat used per cycle = $\cdot 06 \times 600 = 36$ B.Th.U.

Work equivalent $= 36 \times 778 = 28{,}000$ ft.-lbs.

Work done per minute by engine $= 25 \times 33{,}000$ ft.-lbs.

Work done per cycle by engine $= \dfrac{25 \times 33{,}000}{90} = 9{,}170$ ft.-lbs.

$\therefore$ Efficiency of engine $= \dfrac{9{,}170}{28{,}000} = \cdot 33$ approx.

$\therefore$ Efficiency ratio to air standard $= \dfrac{\cdot 33}{\cdot 52} = \cdot 63$ approx.

Effect of Variable Specific Heat of Gas. Up to the present we have assumed that the specific heat of the gas is constant; this, in fact, is now generally admitted not to be the case, and so new formulæ have been devised for the efficiency of the constant volume cycle for variable specific heats.

We may assume that the specific heat C_v of a gas at constant volume increases with increase of temperature according to a linear relation and write

$$C_v = C_{v0} + bt \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1a)$$

Or, for use with absolute temperatures

$$C_v = \beta + b\tau \quad . \quad . \quad . \quad . \quad . \quad . \quad (1b)$$

If, therefore, the absolute temperatures at the points A, B, C, D in the indicator diagram on p. 19 are respectively τ_A, τ_B, τ_C, and τ_D, we have

Heat absorbed from D to A $= H_1 = C_{v1}(\tau_A - \tau_D)$ (2)

,, rejected ,, B to C $= H_2 = C_{v2}(\tau_B - \tau_C)$. (3)

Taking for the specific heats the mean over the interval, we shall have

$$C_{v1} = \beta + \frac{b}{2}(\tau_A + \tau_D) \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$$C_{v2} = \beta + \frac{b}{2}(\tau_B + \tau_C) \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Now, the efficiency of the cycle $= \eta_s = \dfrac{H_1 - H_2}{H_1}$.

$$\eta_s = \frac{(\tau_A - \tau_D)\left\{\beta + \frac{b}{2}(\tau_A + \tau_D)\right\} - (\tau_B - \tau_C)\left\{\beta + \frac{b}{2}(\tau_B + \tau_C)\right\}}{(\tau_A - \tau_D)\left\{\beta + \frac{b}{2}(\tau_A + \tau_D)\right\}} \quad (6)$$

Numerical Value of Specific Heat. Different observers have obtained considerable divergence in the values of the specific heat of gases at constant volume. The following were published by Mallard and Le Chatelier in 1887:—

$$\text{For } CO_2, \quad . \quad C_v = 0{\cdot}1477 + 0{\cdot}176 \frac{t}{1{,}000}.$$

$$H_2O, \quad . \quad . \quad C_v = 0{\cdot}3211 + 0{\cdot}219 \frac{t}{1{,}000}.$$

$$N_2, \quad . \quad . \quad C_v = 0{\cdot}170 + {\cdot}087 \frac{t}{1{,}000}.$$

$$O_2, \quad . \quad C_v = 0{\cdot}1488 + {\cdot}076 \frac{t}{1{,}000}. \quad . \quad . \quad (1)$$

where t = temperature Centigrade.

Dugald Clerk made many important experiments, and found that the curve showing the relation between specific heat and temperature was more nearly a parabola than a straight line, but for a mixture of 1 of gas to 9 of air, the straight portion of Clerk's curve gives

$$C_v = 0{\cdot}194 + {\cdot}051 \frac{t}{1{,}000}. \quad . \quad . \quad (2)$$

For this mixture Burstall gives

$$C_v = 0{\cdot}178 + 0{\cdot}105 \frac{t}{1{,}000}. \quad . \quad . \quad (3)$$

The Gas Explosions Committee of the British Association recommended in their report of 1908 the use of the formula—

$$C_v = 0{\cdot}172 + {\cdot}075 \frac{t}{1{,}000}.$$

Effect of Variable Specific Heat upon Adiabatic Expansion.—Suppose that a gas at pressure p, volume V, and absolute temperature τ has its temperature increased by an amount $d\,\tau$, and its volume decreased by an amount d V.

Then the work done on the gas, expressed in heat units

$$= C_v \; d\,\tau + P \frac{p\,d\,V}{J}$$

But this work will be represented by an increase $d\,\mathrm{H}$ in the internal heat of the gas, so that we have—

$$d\,\mathrm{H} = \mathrm{C}_v\,d\,\tau + \frac{p\,d\,\mathrm{V}}{\mathrm{J}}. \qquad (1)$$

If the process is adiabatic, we must have no increase in the internal heat, so that $d\,\mathrm{H} = 0$, or

$$\mathrm{C}_v\,d\,\tau = -\frac{p\,d\,\mathrm{V}}{\mathrm{J}}. \qquad (2)$$

Now, we have by equation (1), p. 4.

$$\frac{p\,\mathrm{V}}{\tau} = c = \mathrm{J}\,(\mathrm{C}_p - \mathrm{C}_v). \qquad (3)$$

$$\therefore \quad \tau = \frac{p\,\mathrm{V}}{c}$$

$$\frac{d\,\tau}{d\,\mathrm{V}} = \frac{1}{c}\left(\mathrm{V}\,\frac{d\,p}{d\,\mathrm{V}} + p\right)$$

$$\therefore \quad d\tau = \frac{1}{c}\,(\mathrm{V}\,d\,p + p\,d\,\mathrm{V}). \qquad (4)$$

∴ Putting this result in (2) we have

$$\frac{\mathrm{C}_v}{\mathrm{J}\,(\mathrm{C}_p - \mathrm{C}_v)}\,.\,(\mathrm{V}\,d\,p + p\,d\,\mathrm{V}) = -\frac{p\,d\,\mathrm{V}}{\mathrm{J}}$$

$$\therefore \quad \frac{\mathrm{C}_v}{(\mathrm{C}_p - \mathrm{C}_v)}\,.\,\mathrm{V}\,d\,p = -p\,d\,\mathrm{V}\left\{1 + \frac{\mathrm{C}_v}{\mathrm{C}_p - \mathrm{C}_v}\right\}.$$

$$\mathrm{C}_v\,\mathrm{V}d\,p = -\mathrm{C}_p\,p\,d\,\mathrm{V}.$$

$$\therefore \qquad \mathrm{C}_v\,\frac{d\,p}{p} = -\mathrm{C}_p\,\frac{d\,\mathrm{V}}{\mathrm{V}}. \qquad (5)$$

Now let $\mathrm{C}_v = \beta + b\,\tau$
,, $\mathrm{C}_p = \alpha + b\,\tau$ (6)

Then equation (5) becomes

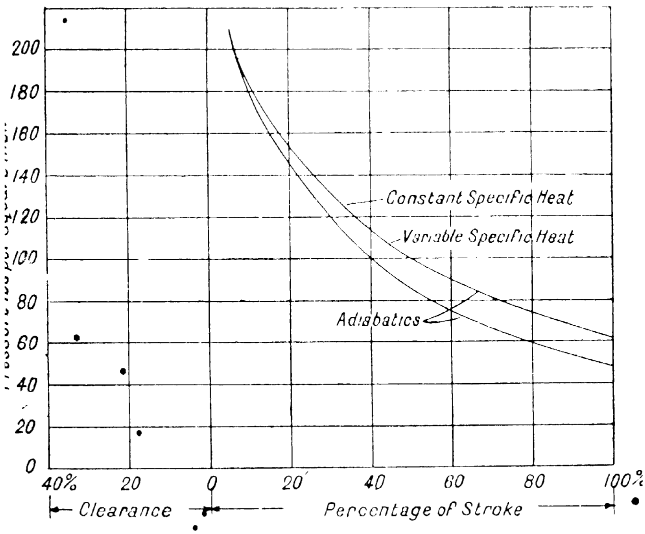
$$(\beta + b\,\tau)\,\frac{d\,p}{p} + (\alpha + b\,\tau)\,.\,\frac{d\,\mathrm{V}}{\mathrm{V}} = 0.$$

$$\beta\,.\,\frac{d\,p}{d} + \alpha\,\frac{d\,\mathrm{V}}{\mathrm{V}} + \frac{b\,\tau}{p\,\mathrm{V}}\,\{\,\mathrm{V}\,d\,p + p\,d\,\mathrm{V}\,\} = 0.$$

i.e.,
$$\beta \frac{dp}{p} + \alpha \frac{dV}{V} + \frac{b}{c} \{ d(pV) \} = 0. \quad . \quad . \quad . \quad (7)$$
$$\left[\text{because } \frac{\tau}{pV} = \frac{1}{c} \right]$$

Integrating, we get
$$\beta \log p + \alpha \log V + \frac{b}{c} pV = \text{constant}$$

i.e.,
$$\beta \log p + \alpha \log V + \frac{bJ}{\alpha - \beta} pV = \text{constant} \quad . \quad (8)$$



ADIABATIC CURVES FOR CONSTANT AND VARIABLE SPECIFIC HEAT

This is the equation for adiabatic expansion of gases the specific heats of which vary in the linear manner given by equations (6).

We note that $\dfrac{V \cdot pV}{(\alpha - \beta)} = \dfrac{pV}{c} = \tau$, so that we may write

equation (8) as :—

$$\beta \log p + \alpha \log V + b\,\tau = \text{constant.} \quad . \quad . \quad . \quad . \quad (9)$$

or, since the absolute temperature is at a fixed interval below the ordinary temperature, we may write the equation as :—

$$\beta \log p + \alpha \log V + b\,t = \text{constant} \quad . \quad . \quad . \quad (10)$$

We will note that if the specific heat is constant, the value of the coefficient b will be zero, and equation (10) becomes

$$\beta \log p + \alpha \log V = \text{constant.}$$

$$i.e., \log p + \frac{\alpha}{\beta} \log V = \text{constant}$$

$$i.e., p\,V^{\frac{\alpha}{\beta}} = \text{constant}$$

But $\dfrac{\alpha}{\beta} = \dfrac{C_p}{C_v} = \gamma$ $\therefore$ we obtain the familiar result $p\,V^{\gamma} = \text{constant}$.

The adiabatic curve for variable specific heat will be found to come above that for constant specific heat, as shown on the accompanying diagram.

Wimperis' Formula for the Ideal Efficiency of the Constant Volume Cycle with Variable Specific Heats. Mr. H. E. Wimperis,* M.A., has given the following useful derivation of a convenient formula for the efficiency of the constant volume cycle with variable specific heats.

Referring back to equation (6), we have for the efficiency of the constant volume cycle with variable specific heat

$$\eta_v = \frac{(\tau_A - \tau_D)\left\{\beta + \frac{b}{2}(\tau_A + \tau_D)\right\} - (\tau_B - \tau_C)\left\{\beta + \frac{b}{2}(\tau_B + \tau_C)\right\}}{(\tau_A - \tau_D)\left\{\beta + \frac{b}{2}(\tau_A + \tau_D)\right\}} \quad (11)$$

Now, take equation (9) and write it :—

$$\log p + \frac{\alpha \log V}{\beta} + \frac{b\,\tau}{\beta} = \text{constant}$$

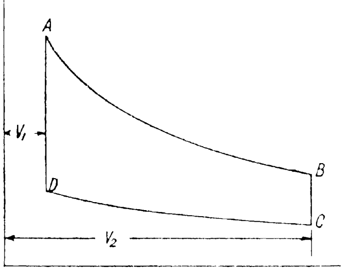
Taking antilogs of each side, we get :—

$$p\,V^{\frac{\alpha}{\beta}}\,\varepsilon^{\frac{b\,\tau}{\beta}} = \text{constant}$$

* *The Internal Combustion Engine* (Constable). Advanced students are recommended to read this interesting and stimulating book.

(ε = exponential coefficient, the base of the Naperian logs), and since $\frac{\alpha}{\beta}$ = ratio of specific heats at absolute zero, we may write $\gamma_0 = \frac{\alpha}{\beta}$, and our equation to the adiabatic expansion with variable specific heats becomes—

$$p\,V^{\gamma_0}\,\varepsilon^{\frac{b\tau}{\beta}} = \text{constant} \qquad (12)$$



Now we have also $\frac{p_A V_1}{\tau_A} = \frac{p_B V_2}{\tau_B}$.

$$i.e.,\ \frac{\tau_A}{\tau_B} = \frac{p_B V_2}{p_A V_1} = \frac{p_B}{p_A}\,r \quad . \quad . \quad . \qquad (13)$$

Equation (12) gives—

$$p_A\,V_A^{\gamma_0}\,\varepsilon^{\frac{b\tau_A}{\beta}} = p_B\,V_B^{\gamma_0}\,\varepsilon^{\frac{b\tau_B}{\beta}}.$$

$$\therefore\ \frac{p_B}{p_A} = \left(\frac{V_A}{V_B}\right)^{\gamma_0}\varepsilon^{\frac{b}{\beta}(\tau_A - \tau_B)} \quad . \quad . \qquad (14)$$

$\therefore$ putting this in (13) we get—

$$\frac{\tau_A}{\tau_B} = r^{1-\gamma_0}\,.\,\varepsilon^{\frac{b}{\beta}(\tau_A - \tau_B)} \quad . \quad . \quad . \quad . \qquad (15a)$$

and similarly we shall get—

$$\frac{\tau_D}{\tau_C} = r^{1-\gamma_0}\,\varepsilon^{\frac{b}{\beta}(\tau_B - \tau_C)} \quad . \quad . \quad . \quad . \qquad (15b)$$

$\therefore$ substituting in equation (11) for τ_B and τ_C we have—

$$\eta_e = \frac{(\tau_A - \tau_D)\left\{\beta + \frac{b}{2}(\tau_A + \tau_D)\right\} - \left\{\tau_A \, . \, r^{1-\gamma_0} \, . \, \varepsilon^{\frac{b}{\beta}(\tau_A - \tau_B)} - \tau_D \, r^{1-\gamma_0} \, \varepsilon^{\frac{b}{\beta}(\tau_D - \tau_C)}\right\}\left\{\beta + b\left(\frac{\tau_B + \tau_C}{2}\right)\right\}}{(\tau_A - \tau_D)\left\{\beta + \frac{b}{2}(\tau_A - \tau_D)\right\}}$$

$$= 1 - \left(\frac{1}{r}\right)^{\gamma_0 - 1}\left[\frac{\left(\tau_A \, \varepsilon^{\frac{b}{\beta}(\tau_A - \tau_B)} - \tau_D \, \varepsilon^{\frac{b}{\beta}(\tau_D - \tau_C)}\right)\left(\beta + b\left(\frac{\tau_B + \tau_C}{2}\right)\right)}{(\tau_A - \tau_D)\left\{\beta + \frac{b}{2}(\tau_A - \tau_D)\right\}}\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

Now $\quad \beta + \frac{b}{2}(\tau_B + \tau_C) = \beta + \frac{b}{2}\left(\tau_A \, r^{1-\gamma_0} \, \varepsilon^{\frac{b}{\beta}(\tau_A - \tau_B)} - \tau_D \, r^{1-\gamma_0} \, \varepsilon^{\frac{b}{\beta}(\tau_D - \tau_C)}\right)$

and taking the first two terms in the expansion of the exponential, the numerator in equation (16) becomes—

$$\left\{\tau_A\left[1 + \frac{b}{\beta}(\tau_A - \tau_B)\right] - \tau_D\left[1 + \frac{b}{\beta}(\tau_D - \tau_C)\right]\right\}\left\{\beta + \frac{b}{2}r^{1-\gamma_0}\left[\tau_A\left(1 + \frac{b}{\beta}(\tau_A - \tau_B)\right) + \tau_D\left(1 + \frac{b}{\beta}(\tau_D - \tau_C)\right)\right]\right\}$$

$$= \left\{\tau_A - \tau_D + \frac{b}{\beta}(\tau_A^2 - \tau_B\tau_A - \tau_D^2 + \tau_D\tau_C)\right\}\left\{\beta + \frac{b}{2}r^{1-\gamma_0}\left[\tau_A + \tau_D + \frac{b}{\beta}(\tau_A^2 - \tau_A\tau_B + \tau_D^2 - \tau_D\tau_C)\right]\right\} \quad . \quad . \quad (17)$$

The left half of this expression may be written to a sufficient degree of approximation—

$$\left\{(\tau_A - \tau_D) + \frac{b}{\beta}(\tau_A{}^2 - \tau_A{}^2 r^{1-\gamma_0} - \tau_D{}^2 - \tau_D{}^2 r^{1-\gamma_0})\right\} = \left\{(\tau_A - \tau_D) - \frac{b}{\beta}(\tau_A{}^2 - \tau_D{}^2)(1 - r^{1-\gamma_0})\right\}$$

$$= (\tau_A - \tau_D)\left\{1 + \frac{b}{\beta}(\tau_A + \tau_D)(1 - r^{1-\gamma_0})\right\} \quad . \quad (18)$$

The square bracket in equation (16) may, therefore, be written—

$$\left[\frac{(\tau_A - \tau_D)\left\{1 + \frac{b}{\beta}(\tau_A + \tau_D)(1 - r^{1-\gamma_0})\right\}\left\{1 - \frac{b}{2\beta}r^{1-\gamma_0}\left[\tau_A - \tau_D - \frac{b}{\beta}(\tau_A{}^2 - \tau_A\tau_D - \tau_D{}^2 - \tau_B\tau_C)\right]\right\}}{(\tau_A - \tau_D)\left\{1 + \frac{b}{\beta}\left(\frac{\tau_A - \tau_D}{2}\right)\right\}}\right]$$

Now expand $\left(\dfrac{1}{1 + \frac{b}{\beta}\left(\frac{\tau_A - \tau_D}{2}\right)}\right)$ and neglect terms involving $\left(\dfrac{b}{\beta}\right)^2$, and the above expression can be reduced as follows—

$$\left\{1 + \frac{b}{\beta}(\tau_A + \tau_D)(1 - r^{1-\gamma_0})\right\}\left\{1 - \frac{b}{2\beta}r^{1-\gamma_0}\left[\tau_A - \tau_D - \frac{b}{\beta}(\tau_A{}^2 - \tau_A\tau_B - \tau_D{}^2 - \tau_B\tau_C)\right]\right\}\left\{1 - \frac{b}{\beta}\left(\frac{\tau_A - \tau_D}{2}\right)\right\}$$

$$= 1 + \frac{b}{\beta}(1 - r^{1-\gamma_0})(\tau_A + \tau_D) - \frac{s}{2\beta}r^{1-\gamma_0}(\tau_A - \tau_D) - \frac{s}{\beta}\left(\frac{\tau_A - \tau_D}{2}\right)$$

$$= 1 + \frac{b}{\beta}\left\{1 - r^{1-\gamma_0}(\tau_A + \tau_D) + \tfrac{1}{2}\, r^{1-\gamma_0}(\tau_A - \tau_D) - \tfrac{1}{2}\,\tau_A - \tfrac{1}{2}\,\tau_D\right\}$$

$$= 1 + \frac{b}{\beta}\left\{\tfrac{1}{2}\,\tau_A + \tfrac{1}{2}\,\tau_D - \tfrac{1}{2}\,\tau_A\, r^{1-\gamma_0} - \tfrac{1}{2}\,\tau_D\, r^{1-\gamma_0}\right\}$$

$$= 1 + \frac{b}{2\beta}\left\{\tau_A\left(1 - r^{1-\gamma_0}\right) + \tau_D\left(1 - r^{1-\gamma_0}\right)\right\} = 1 + \frac{b}{2\beta}(\tau_A + \tau_D)\left(1 - r^{1-\gamma_0}\right) \qquad (19)$$

$\therefore$ Going back to equation (16) we see that—

$$\eta_s = 1 - \left(\frac{1}{r}\right)^{\gamma_0 - 1}\left[1 + \frac{b}{2\beta}(\tau_A + \tau_D)\left(1 - r^{1-\gamma_0}\right)\right] \quad . \quad . \qquad (20)$$

but by equation (7), p. 50,

$$1 - \left(\frac{1}{r}\right)^{\gamma_0 - 1} = \eta = \text{efficiency with constant specific heat}$$

$$\therefore \qquad \eta_s = \eta - \left(\frac{1}{r}\right)^{\gamma_0 - 1}\frac{b}{2\beta}\,\eta\,(\tau_A + \tau_D) = \eta\left\{1 - \left(\frac{1}{r}\right)^{\gamma_0 - 1}\frac{b}{2\beta}(\tau_A + \tau_D)\right\}$$

$$= \eta\left\{1 - (1 - \eta)\,\frac{b}{\beta}\left(\frac{\tau_A + \tau_D}{2}\right)\right\} \quad . \quad . \quad . \quad . \quad . \quad . \qquad (21)$$

The compression temperature τ_D is not easy to determine in practice, and as an approximation, since τ_D is multiplied by the small quantity $\frac{b}{\beta}$, we may write

$$\frac{\tau_D}{\tau_C} = r^{\gamma - 1} = \frac{1}{1 - \eta_i}.$$

$$\therefore \qquad \eta_v = \eta_i\left\{1 - (1 - \eta_i)\frac{b}{2\beta}\left(\tau_A + \frac{\tau_C}{(1 - \eta_i)}\right)\right\}$$

$$= \eta_i\left[1 - \frac{b}{2\beta}\left\{(1 - \eta_i)\tau_A + \tau_C\right\}\right] \qquad (22)$$

This is Wimpens' formula

Comparison of Ideal Efficiencies for Constant and Variable Specific Heats. The following table shows the effect upon the ideal efficiency of the allowance for variation of the specific heat of gas with temperature

Expansion ratio r	Efficiency Air Standard η_i	Efficiency for Variable Specific Heat η_v	Ratio $\frac{\eta_v}{\eta_i}$
4	0.426	0.335	0.787
5	0.475	0.384	0.809
6	0.512	0.417	0.815
7	0.541	0.445	0.823
8	0.565	0.470	0.832
9	0.585	0.490	0.837
10	0.602	0.508	0.835

CONSTANT PRESSURE CYCLE

This cycle was the basis of the Ericsson air engine and the Joule air engine in the former the expansion and compression were isothermal, and in the latter they are more nearly adiabatic The Joule cycle has been revived in recent years in the Diesel engine, which approaches this cycle

Restricting our attention to the Joule cycle, which is the only one now used in practice, we have the following cycle

(1) Heat taken in at constant pressure from A to B, during this stage the fuel burns

(2) Gases expand adiabatically from B to C.

(3) Gases expelled at constant volume to D and new charge of fuel taken in. in the four-stroke cycle * the gases are expelled by an idle stroke of the piston to E, and a new charge of fuel is taken in by an idle outward stroke of the piston to C, which returns to D before compression begins

(4) Adiabatic compression from D to A

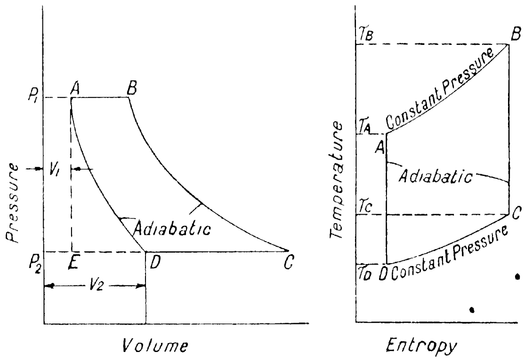
Efficiency of Constant Pressure Cycle

Heat supplied at constant pressure from A to B

$$C_p(\tau_B - \tau_A) \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Heat rejected at constant pressure from C to D

$$C_p(\tau_C - \tau_D) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$



Heat converted into work = heat supplied − heat rejected

$$= C_p\{(\tau_B - \tau_A) - (\tau_C - \tau_D)\}$$

$$\therefore \quad \text{Efficiency} = \eta = \frac{\text{heat converted into work}}{\text{heat supplied}}$$

$$= \frac{(\tau_B - \tau_A) - (\tau_C - \tau_D)}{(\tau_B - \tau_A)}$$

$$= 1 - \frac{(\tau_C - \tau_D)}{(\tau_B - \tau_A)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

* See p. 143.

But by equation (10), p. 11.

$$\frac{\tau_A}{\tau_D} = \frac{\tau_B}{\tau_C} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} \quad . \quad . \qquad (4)$$

$$\therefore \quad \frac{\tau_A}{\tau_B} = \frac{\tau_D}{\tau_C}$$

$$\therefore \quad \frac{\tau_A}{\tau_B - \tau_A} = \frac{\tau_D}{\tau_C - \tau_D}$$

$$i.e., \quad \frac{\tau_C - \tau_D}{\tau_B - \tau_A} = \frac{\tau_D}{\tau_A}.$$

$$\therefore \quad \eta = 1 - \frac{\tau_D}{\tau_A} = 1 - \frac{\tau_C}{\tau_B} \quad . \quad . \qquad (5)$$

$$= 1 - \left(\frac{1}{r}\right)^{\gamma-1} \quad . \quad . \quad . \qquad (6)$$

where, as before, r = ratio of compression $= \frac{V_2}{V_1}$.

The efficiency of this cycle is less than that of a perfect engine working between the temperature limits, because all the heat is not taken in and rejected at the same temperature.

RANKINE-CLAUSIUS CYCLE

This cycle is sometimes called the Rankine cycle, and was adopted by the 1898 Report of the Institution of Civil Engineers on the Thermal Efficiency of Steam Engines as the standard of comparison with the performance of a steam engine working under the same conditions.

We thus obtain a quantity which may be called the Rankine efficiency coefficient, and may be defined as follows:

Rankine efficiency coefficient

$$= R_\eta = \frac{\text{thermal efficiency of actual engine}}{\text{thermal efficiency of ideal engine on Rankine cycle}}$$

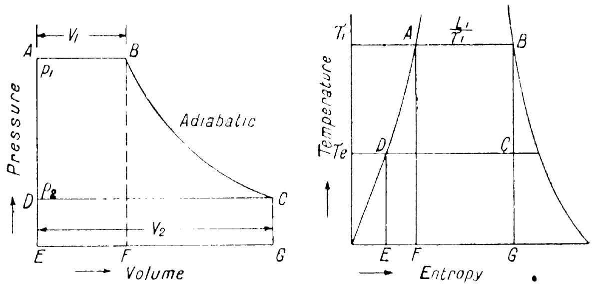
Nature of Cycle.—The p V and $\tau\varphi$ diagrams are given on the accompanying diagram, and the cycle comprises the following stages:—

(1) Steam at pressure p_1 is drawn into the cylinder at constant temperature from A to B, the volume at B being V_1; or, expressed

in other terms, water at pressure p_1 and temperature τ_i (the volume of which is negligible) represented by A is converted into dry saturated steam at constant pressure, the volume of steam being V_1. The corresponding line A B of the $\tau\phi$ diagram will be the horizontal line A B, the length of which is $\frac{L_i}{\tau_i}$.

(2) The steam expands adiabatically until its pressure is p_2, volume V_2, and temperature τ_e. Corresponding to this we have the curve B C on the p V diagram and the vertical line B C on the $\tau\phi$ diagram.

(3) The steam is condensed to water at constant pressure and temperature. This stage is represented by C D on the p V, and



$\tau\phi$ diagrams, the line being a horizontal straight line on both diagrams, because pressure and temperature are constant. Further, the point D must be on the water boundary curve of the $\tau\phi$ diagram, because by the time the point D is reached on the p V curve, all the steam has become condensed to water.

(4) The water is pumped back into the boiler, its pressure being increased to p_1 and its temperature to τ_i: this is represented by D A on both diagrams.

Efficiency on Rankine-Clausius without Superheat. (1) *From Consideration of p V Diagram.* If H_i is the total heat of steam at the inlet temperature, and s_e is the sensible heat at the exhaust temperature, we have

Heat supplied per stroke $= J\,(H_i - s_e)$ in mechanical units.

The work done per stroke = area A B C D

$$= \text{area A B F E} + \text{area B C G F} - \text{area D C G E}$$

$$= p_1 V_1 + \frac{p_1 V_1 - p_2 V_2}{\gamma - 1} - p_2 V_2$$

$$= (p_1 V_1 - p_2 V_2)\left\{1 + \frac{1}{\gamma - 1}\right\}$$

$$= \frac{\gamma}{\gamma - 1}\{p_1 V_1 - p_2 V_2\}. \quad . \quad . \quad (1)$$

Or, if r is the expansion ratio $\frac{V_2}{V_1}$

$$\text{Work per stroke} = \frac{\gamma V_1}{(\gamma - 1)}\{p_1 - r p_2\}. \quad . \quad . \quad (2)$$

$$\therefore \quad \text{Efficiency} \quad \eta = \frac{\text{work per stroke}}{\text{heat supplied}}$$

$$= \frac{\gamma V_1 (p_1 - r p_2)}{(\gamma - 1) J (H_t - s_e)}. \quad . \quad . \quad (3)$$

Zeuner gives the following value for γ, and this is generally taken as the most reliable, viz.:—

$$\gamma = 1{\cdot}035 + {\cdot}1\, x,$$

where x is the dryness fraction of the steam.

In our case we have assumed the steam to be dry saturated, so that $x = 1$ and $\gamma = 1{\cdot}135$.

$\therefore$ Equation (3) becomes—

$$\eta = \frac{1{\cdot}135\, V_1 (p_1 - r p_2)}{{\cdot}135\, J\, (H_t - s_e)}. \quad . \quad . \quad . \quad . \quad (4)$$

If pressures are in pounds per square inch, volumes in cubic feet, and H_t and s_e are in B.Th.U., J = 778 ft.-lbs per B.Th.U., and we have to multiply the pressures by 144 to bring them to lbs. per square feet to preserve the dimensions of the equation.

[This will be clear from the following dimensional equation, in which the numerical values are neglected.

$$\eta = \text{ft.}^2 \times \text{in.}^2 = \frac{\text{ft}^2}{\text{in.}^2}$$

Multiplying by 144 brings this to $\frac{\text{in}^2}{\text{in.}^2} = 1$, giving η as a dimensionless coefficient]

We then obtain

$$\eta = \frac{1{\cdot}55\, V_1 (p_1 - r\, p_2)}{(H_1 - s_e)}.$$

In the above treatment we have neglected the work done in pumping the condensed water into the boiler. If Vw is the volume of the water, the work thus spent is $Vw\,(p_1 - p_2)$, and this should strictly be deducted from the heat converted into work, but for most practical calculations this may be neglected.

Numerical Example. *The inlet pressure in a steam engine is* 100 *lbs per square inch absolute* ($t = 328°$ *F.*), *and the exhaust pressure is* 14·7 *lbs per square inch* ($t = 212°$ *F.*), *the ratio of expansion being* 5·1. *If there are* 1·31 *cubic feet of steam per pound at the inlet pressure, what will be the efficiency on the Rankine-Clausius cycle?*

Taking from tables $H_1 = 1{,}180$, we have—

$$\eta = \frac{1{\cdot}55 \times 1{\cdot}31\,(100 - 5{\cdot}1 \times 14{\cdot}7)}{1{,}180 - (212 - 32)}$$

$$= \frac{1{\cdot}55 \times 1{\cdot}31 \times 20{\cdot}6}{1{,}000} = {\cdot}139$$

$$= \mathbf{13{\cdot}9} \text{ per cent.}$$

(2) *From Consideration of $\tau\varphi$ Diagram.* The work done per pound of steam is represented by the area A B C D, and the heat supplied is represented by the area E D A B G, and is equal to $(H_i - s_e)$ or $(L_i + \tau_i - \tau_e)$.

Now the area A B C D = area A B C H + area D A H

$$= \frac{L_i}{\tau_i}(\tau_i - \tau_e) + \text{area E D A F} - \text{area E D H F}.$$

But area E D A F = increase in heat of water from τ_e to τ_i

$$= (\tau_i - \tau_e).$$

Area E D H F = E D × D H

$$= \tau_e \log_e \frac{\tau_1}{\tau_e}.$$

$\therefore$ Work done $= \frac{L_1}{\tau_1}(\tau_1 - \tau_e) + (\tau_1 - \tau_e) - \tau_e \log_e \frac{\tau_1}{\tau_e}.$

$$= (\tau_1 - \tau_e)\left(1 + \frac{L_1}{\tau_1}\right) - \tau_e \log_e \frac{\tau_1}{\tau_e}. \quad (1)$$

$\therefore$ Efficiency $= \eta = \frac{\text{work done}}{\text{work supplied}}$

$$= \frac{(\tau_1 - \tau_e)\left(1 + \frac{L_1}{\tau_1}\right) - \tau_e \log_e \frac{\tau_1}{\tau_e}}{L_1 + \tau_1 - \tau_e}. \quad (2)$$

For approximate calculations, in which it is inconvenient to calculate by hyperbolic logarithms, the work done may be obtained from a $\tau\phi$ chart for steam, such as that reproduced on p. 23, as follows:

Measure A B and D J on the diagram.

Then work done $= (\tau_1 - \tau_e)(1 + \text{A B}) - \tau_e \,.\, \text{D J}.$

Numerical Example. Take the same example as worked on p. 66, and determine the efficiency of the cycle by means of the entropy formula.

We have, as before, $L_1 + \tau_1 - \tau_e = H_1 - s_e = 1{,}000$ B.Th.U.

$$L_1 = 1{,}180 - (328 - 32) = 884.$$

$$\tau_1 - \tau_e = 328 - 212 = 116.$$

$$\tau_1 = 328 + 461 = 789.$$

$$\tau_e = 212 + 461 = 673.$$

$$1 + \frac{L_1}{\tau_1} = 1 + \frac{884}{789} = 2{\cdot}12.$$

$$\log_e \frac{\tau_1}{\tau_e} = 2{\cdot}3\,(\log 789 - \log 673) = {\cdot}159.$$

$$\therefore \quad \eta = \frac{116 \times 2{\cdot}12 - 673 \times {\cdot}159}{1{,}000} = {\cdot}140$$

$$= \mathbf{14{\cdot}0} \text{ per cent.}$$

This agrees with the previous result to the degree of accuracy which is possible in calculations of this kind.

Comparison with Carnot Efficiency. The Rankine-Clausius cycle is not a reversible one, so that its efficiency must be less than for the Carnot cycle for the same temperature limits.

In the numerical example which we have just considered, we should have

$$\text{Carnot efficiency} = \frac{\tau_i - \tau_e}{\tau_i} = \frac{116}{789} = \cdot 147 = \mathbf{14{\cdot}7} \text{ per cent.}$$

In view of the very simple calculation required for the Carnot efficiency, and of the fact that all calculations on the thermal efficiency of heat engines are necessarily only approximate, it appears doubtful whether the Rankine-Clausius is a better ideal standard of comparison than the Carnot. In practice the actual thermal efficiency seldom exceeds 60 per cent. of the Rankine efficiency, and it is often very much less.

Rankine-Clausius Cycle with Superheated Steam.—If the temperature of superheat is t_s and the specific heat of superheated steam is k (see p. 57, Vol. I., for values of k), we can calculate the efficiency of a steam engine working upon the Rankine-Clausius cycle by means of the $\tau\varphi$ diagram, as follows:

$$\left.\begin{matrix}\text{Heat supplied per}\\ \text{lb. of steam}\end{matrix}\right\} = \mathrm{L}_i + \tau_i - \tau_e + k\,t_s \quad . \quad . \quad . \quad (1)$$

$$\left.\begin{matrix}\text{Work done per lb.}\\ \text{of steam}\end{matrix}\right\} = \text{area A B H K D}$$

$$= \text{area A B C D} + \text{area B H K C} \quad . \quad . \quad (2)$$

We have already shown on p. 67 that—

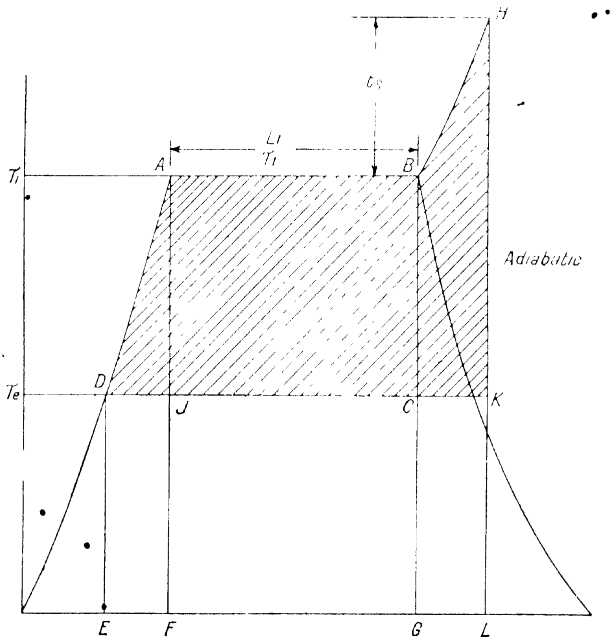
$$\text{Area A B C D} = (\tau_i - \tau_e)\left(1 + \frac{\mathrm{L}_i}{\tau_i}\right) - \tau_e \log_e \frac{\tau_i}{\tau_e} \quad . \quad . \quad (3)$$

Now area B H K C = area G B H L − area G C K L

$$= k\,t_s - \tau_e\,k \log_e \frac{\tau_i + t_s}{\tau_i} \quad . \quad . \quad . \quad (4)$$

Because G B H L represents the total increase of heat in passing from dry saturated steam at τ_i to superheated steam at $(\tau_i + t_s)$

= *i.e.*, $k \cdot t_s$, and GCKL = GC + CK, and CK = increase in entropy in passing from B to H = *i.e.*, $k \log_e \frac{\tau_1 + t_s}{\tau_1}$.



∴ Work done per lb. of steam

$$= (\tau_1 - \tau_e)\left(1 + \frac{L_1}{\tau_1}\right) + k\,t_s - \tau_e\left\{\log_e \frac{\tau_1}{\tau_e} + k \log_e\left(\frac{\tau_1 + t_s}{\tau_1}\right)\right\} \quad (5)$$

∴ Efficiency of cycle

$$= \eta = \frac{(\tau_1 - \tau_e)\left(1 + \frac{L_1}{T_1}\right) + k\,t_s - \tau_e\left\{\log_e \frac{\tau_1}{\tau_e} + k \log_e\left(\frac{\tau_1 + t_s}{T_1}\right)\right\}}{L_1 + \tau_1 - \tau_e + k\,t_s}. \quad (6)$$

LECTURE IV.—QUESTIONS.

1. Explain the cycle used in Stirling's regenerator air engine, showing an ideal indicator card.

2. Show that on theoretical grounds the efficiency of an ideal gas engine working between temperatures T_2, T_1, and T_0 of exhaust, explosion, and compression respectively is

$$1 - \frac{T_2}{T_1 - T_0} \log_e \frac{T_1}{T_0}$$

3. An engine compressing to 240 lbs. per square inch turns into work 38 per cent of the heat supplied. Compare this with an efficiency of a similar engine working with constant specific heats, all heat being added at constant volume and expansion, and compression being adiabatic. (V = 1·41.)

4. In an ideal air engine receiving heat at constant pressure (specific heat 0·2375) and rejecting heat at constant volume (specific heat 0·169), the expansion and compression are adiabatic, and the ratio of compression is 15 and of expansion is 7. The temperature at the beginning of compression is 104° F. (72° C.). The pressure at the beginning of compression is 15 and at release 45 lbs. per square inch absolute. Find the work done per pound of working substance. Draw the $\tau - \phi$ chart, finding three points on the heat admission and heat rejection lines.

5. Calculate from the steam tables the maximum work obtainable from 1 lb. of dry saturated steam when working between pressures of 100 and 2 lbs. per square inch absolute (*a*) on the Rankine cycle, (*b*) on the Carnot cycle. Account for the difference obtained.

6. Find an expression for the efficiency of an ideal engine working on the Otto cycle, assuming air as the working substance, and the specific heat constant. Compare on a temperature basis the efficiency of an engine working on the Carnot cycle with an engine working on the Otto cycle, between the same limits of temperature.

7. Superheated steam at a pressure of 150 lbs. per square inch absolute and temperature 458° F. (236·7° C.) expands adiabatically to 3 lbs. per square inch absolute; assuming the Rankine cycle is followed by this steam, determine the pounds of steam required per hour per horse-power. What is the dryness of the steam after expansion?

Lecture IV. A.M.Inst.C.E. Questions

1. The following results were obtained from the test of a gas engine working on the Otto cycle:—

Duration of test in minutes, . .	90
Indicated horse-power,	150
Total gas used in cubic feet,	3,825
Calorific value of gas per cubic foot, B.Th.U.,	550
Clearance volume, 15 per cent. of piston displacement	

Find the thermal efficiency and the efficiency-ratio of the engine.

2. Find the work done per pound of steam by a steam engine working on the Rankine cycle between 362° F. and 152° F. How many pounds of steam are required per horse-power, given the following table?

Temperature °F.	Entropy of 1 lb. Water	Total Entropy of 1 lb. of Steam
362	0.519	1.566
152	0.218	1.863

3. Use the following table to draw a temperature-entropy diagram, and from it find the work done per pound of dry steam by a perfect engine working on the Rankine cycle between 150 lbs. per square inch and 16 lbs. per square inch absolute pressures. Also determine the dryness of the steam after expansion.

Pressure lbs. per Square Inch	Temperature F.	Entropy of 1 lb. of Water	Total Entropy of 1 lb. of Steam
150	359	0.514	1.569
16	216	0.319	1.749

4. What is the difference between the theoretical Otto and Diesel cycles? Which is theoretically the most efficient, assuming the same ratio of compression in both cases? Which is practically the most efficient, and why?

5. Assuming constant specific heat, sketch the theoretical Otto cycle both on the pv and on the $\tau\phi$ diagram, and explain the meaning of each line. Superimpose in each case diagrams that might be obtained from an actual engine, stating the causes of the differences between the actual and the theoretical diagrams.

6. Explain the terms specific heat at constant volume, and specific heat at constant pressure. What views are now held as to the variation of these specific heats with temperature and with pressure in respect of the products of combustion of a gas engine?

7. In an ideal engine cycle with constant admission and back pressure and curves of expansion and compression pv^n = constant, show that the amount of clearance has no effect upon the efficiency, provided the expansion is carried to the lower and the compression to the higher pressure. What are the effects of clearance and compression in ordinary engines?

8. In an engine working on the Rankine-Clausius cycle, the steam being dry saturated at entry, the limits of temperature are 344° F. (latent heat 880 B.Th.U.) and 126° F. (latent heat 1,020 B.Th.U.), the thermal efficiency is 25 per cent. If the steam is superheated 200° F. at inlet, calculate the additional work done and the thermal efficiency. Explain why the addition of heat as superheat gives proportionately more work (1) in theory, (2) in practice.

9. In a gas engine working on the Otto cycle the clearance volume is one-third of the volume swept by the piston per stroke. Find the thermal efficiency of the corresponding ideal air cycle. Find also the mean pressure in the ideal cycle if the pressure at the beginning of compression is 15 lbs. per square inch and if the maximum pressure is three times the pressure at the end of compression. Take the ratio of specific heats as 1.4.

10. Find an expression for the work done per pound of steam by an engine working on the Rankine cycle. An engine working on this cycle receives steam at 370° F. and rejects it at 120° F., and uses 16 lbs. of steam per hour per indicated horse-power. Compare the efficiency of the engine with that of a theoretically perfect engine, working on the same cycle and between the same temperatures.

11. Find the thermal efficiency of the Carnot cycle, and also of the Rankine engine for the range of temperature of T_1 to T_2. What is meant by the term "efficiency ratio"?

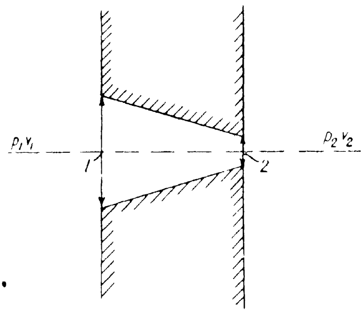
12. Prove that the thermal efficiency of the constant volume gas-engine cycle is $\frac{T_c - T_a}{T_c}$, where T_a is the absolute temperature of the charge before compression, and T_c is its absolute temperature of the charge after compression.

LECTURE V.

FLOW OF STEAM AND GAS.

CONTENTS. —Flow of Gas or Steam through a Nozzle or Orifice—Flow of Steam through a Nozzle or Orifice considered from the Total Heat of the Steam—Ratio of Pressures for Maximum Delivery—Maximum Discharge and Velocity for Steam—Numerical Examples—Design of Steam Nozzles—Flow of Steam through Pipes—Tabulated Values—Losses due to Elbows, Valves, etc.—Numerical Example—Flow of Air through Pipes—Questions.

Flow of Gas or Steam through a Nozzle or Orifice. Suppose that steam or gas flows from one section 1 at pressure p_1 to a second section 2 at lower pressure p_2 and suppose that the flow



is adiabatic. Then the work done in the adiabatic expansion must all be expended in increasing the kinetic energy of the fluid so that if v_1 and v_2 represent the velocities at the sections 1 and 2 so that if a unit quantity passes, having volumes V_1 at 1 and V_2 at 2, we shall have:—

$$\text{Increase in kinetic energy} = \frac{v_2^2 - v_1^2}{2g}. \quad . \quad . \quad (1)$$

Each unit quantity entering section 1 has work done on it by the fluid behind equal to $p_1 V_1$, and in passing out of space 2

it does work on the fluid in front of it equal to $p_2 V_2$, and the work done in the actual adiabatic expansion (see equation (6*a*), p. 11) is equal to $\dfrac{p_1 V_1 - p_2 V_2}{\gamma - 1}$.

$\therefore$ Increase in kinetic energy = total work done on fluid

$$= p_1 V_1 - p_2 V_2 + \frac{p_1 V_1 - p_2 V_2}{(\gamma - 1)}$$

$$= \left(\frac{\gamma}{\gamma - 1}\right)(p_1 V_1 - p_2 V_2). \quad (2)$$

But in adiabatic expansion,

$$p_1 V_1^{\gamma} = p_2 V_2^{\gamma}$$

$$\therefore \qquad V_2 = V_1 \left(\frac{p_1}{p_2}\right)^{\frac{1}{\gamma}}$$

i.e.,

$$p_1 V_1 - p_2 V_2 = p_1 V_1 - p_2 V_1 \left(\frac{p_1}{p_2}\right)^{\frac{1}{\gamma}}$$

$$= p_1 V_1 \left\{1 - \frac{p_2}{p_1}\left(\frac{p_1}{p_2}\right)^{\frac{1}{\gamma}}\right\}$$

$$= p_1 V_1 \left\{1 - \left(\frac{p_2}{p_1}\right)^{1 - \frac{1}{\gamma}}\right\}$$

We, therefore, have

$$\frac{v_2^2 - v_1^2}{2g} = \frac{\gamma}{(\gamma - 1)} \cdot p_1 V_1 \left\{1 - \left(\frac{p_2}{p_1}\right)^{1 - \frac{1}{\gamma}}\right\}. \quad . \quad (3)$$

This is the general equation for the adiabatic flow of a gas, and it makes no allowance for loss of energy due to friction.

In the ordinary case of nozzles or orifices arising in practice, the velocity v_1, in the container in which the pressure is p_1, is negligible, so that we have for the outlet velocity of the jet—

$$v_2 = \sqrt{\frac{2g\gamma}{(\gamma - 1)} \cdot p_1 V_1 \left\{1 - \left(\frac{p_2}{p_1}\right)^{1 - \frac{1}{\gamma}}\right\}}. \quad . \quad (4)$$

Flow of Steam through a Nozzle or Orifice considered from the Total Heat of the Steam. The velocity of steam passing through a nozzle can also be ascertained from a consideration of its loss of total heat, assuming adiabatic expansion. If H_1 is the total heat in thermal units per pound of the steam before it passes into the nozzle, and H_2 is the total heat as it emerges, then if the initial velocity is negligible, we shall have

$$\frac{v_2^2}{2g} = J(H_1 - H_2). \qquad \ldots \ldots \ldots \qquad (5)$$

The values of H_1 and H_2 can be conveniently obtained by means of the Mollier diagram; otherwise we may work as follows, taking x_1 and x_2 as the dryness coefficient of the steam before and after expansion—

$$H_1 = s_1 + x_1 L_1$$

$$H_2 = s_2 + x_2 L_2$$

$$\therefore \quad \frac{v_2^2}{2g} = J\{s_1 - s_2 + x_1 L_1 - x_2 L_2\} \qquad \ldots \qquad (6)$$

x_2 can be obtained by means of the $\tau\phi$ chart for steam in the manner indicated by the diagram on p. 25.

Or if τ_1 and τ_2 are the corresponding temperatures we shall have $s_1 - s_2 = \tau_1 - \tau_2$, and by equation (8), p. 25,

$$\frac{x_2 L_2}{\tau_2} = \log_e \frac{\tau_1}{\tau_2} + \frac{x_1 L_1}{\tau_1}.$$

From this we can calculate x_2, or substituting in equation (6), we shall have—

$$\frac{v_2^2}{2g} = J\left\{\tau_1 - \tau_2 + x_1 L_1 - \tau_2 \log_e \frac{\tau_1}{\tau_2} - \tau_2 \frac{x_1 L_1}{\tau_1}\right\}$$

$$= J\left\{(\tau_1 - \tau_2)\left(1 + \frac{x_1 L_1}{\tau_1}\right) - \tau_2 \log_e \frac{\tau_1}{\tau_2}\right\}. \qquad \ldots \qquad (7)$$

Ratio of Pressures for Maximum Delivery.—If A is the area of the orifice at the point where the velocity is v_2, the weight W discharged per unit of time will be given by the relation

$$W = \frac{\text{volume discharged per unit time}}{\text{volume of unit weight}} = \frac{A v_2}{V_2}.$$

Moreover, $$V_2 = V_1 \left(\frac{p_1}{p_2}\right)^{\frac{1}{\gamma}}.$$

$\therefore$ From equation (1)

$$W = \frac{A}{V_1 \left(\frac{p_1}{p_2}\right)^{\frac{1}{\gamma}}} \sqrt{\frac{2g\gamma}{(\gamma - 1)} p_1 V_1 \left\{ 1 - \left(\frac{p_2}{p_1}\right)^{1 - \frac{1}{\gamma}} \right\}}$$

$$= A \sqrt{\frac{2g\gamma}{(\gamma - 1)} \left(\frac{p_2}{p_1}\right)^{\frac{2}{\gamma}} \frac{p_1 V_1}{V_1^2} \left\{ 1 - \left(\frac{p_2}{p_1}\right)^{1 - \frac{1}{\gamma}} \right\}}$$

$$= A \sqrt{\frac{2g\gamma}{(\gamma - 1)} \frac{p_1}{V_1} \left\{ \left(\frac{p_2}{p_1}\right)^{\frac{2}{\gamma}} - \left(\frac{p_2}{p_1}\right)^{1 + \frac{1}{\gamma}} \right\}}. \quad (8)$$

This will be a maximum when—

$$\left(\frac{p_2}{p_1}\right)^{\frac{2}{\gamma}} - \left(\frac{p_2}{p_1}\right)^{1 + \frac{1}{\gamma}} \text{ is a maximum.}$$

If we write $\left(\frac{p_2}{p_1}\right)$ = pressure ratio = r_p, we shall have a maximum flow when $r_p^{\frac{2}{\gamma}} - r_p^{1 + \frac{1}{\gamma}}$ is a maximum.

i.e., when $$\frac{d\left(r_p^{\frac{2}{\gamma}} - r_p^{1 + \frac{1}{\gamma}}\right)}{d r_p} = 0.$$

i.e., when $$\frac{2}{\gamma} r_p^{\frac{2}{\gamma} - 1} - \left(1 + \frac{1}{\gamma}\right) r_p^{\frac{1}{\gamma}} = 0$$

i.e., when $$\frac{r_p^{\frac{2}{\gamma} - 1}}{r_p^{\frac{1}{\gamma}}} = \frac{1 + \frac{1}{\gamma}}{\frac{2}{\gamma}} = \frac{\gamma + 1}{2}$$

i.e., when $$r_p^{\frac{1}{\gamma} - 1} = \frac{\gamma + 1}{2}.$$

$$i.e., \text{ when } \qquad r_p = \left(\frac{\gamma+1}{2}\right)^{\frac{1}{\frac{1}{\gamma}-1}} = \left(\frac{\gamma+1}{2}\right)^{\frac{\gamma}{1-\gamma}}$$

$$= \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

Maximum Discharge and Velocity for Steam. For dry saturated steam we may take $\gamma = 1 \cdot 135$, and equation (9) gives—

$$r_p = \cdot 577.$$

Substituting in equation (8) for this value, and taking A in square feet, p_1 in lbs. per square inch, and V_1 as cubic feet per pound, we get—

$$W \text{ (in pounds per second)} = 43\,A\sqrt{\frac{V_1}{p_1}}. \quad . \quad . \quad (10)$$

Or, if we take the area in square inches and the flow in pounds per minute, we shall have—

$$\text{Max flow of steam in pounds per minute} = 18\,A\sqrt{\frac{V_1}{p_1}} \qquad (13)$$

Rankine's approximate formula gave

$$\text{Flow in pounds per minute} = \frac{6\,A\,p_1}{7}. \quad . \quad . \quad (14)$$

Substituting in equation (4) the above values of γ and $\frac{p_2}{p_1}$, we shall have

$$v_{2\,max} = 70\sqrt{p_1 V_1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

Numerical Examples. (1) *Plot a diagram showing the maximum flow of steam in pounds per minute passing through an orifice 1 square inch in area) from a space the pressure in which varies from 50 to 150 lbs. per square inch*

In this case the area is 1 square inch, so that A = 1. We will, therefore, tabulate as follows:—

Pressure of steam, lbs./ins.^{2}, p_1,	50	60	70	80	90	100	110	120	130	140	150
Volume of steam per lb. (cubic ft.) V_1, ...	8·42	7·10	6·14	5·43	4·86	4·40	4·03	3·71	3·44	3·21	3·01
Max. flow in lbs. per minute [formula 13], . . .	43·9	52·3	60·7	69·1	77·5	85·8	94·0	102·3	110·6	118·9	127·1
Approx. max. flow in lbs. per minute by Rankine's formula,	42·9	51·3	60·0	68·6	77·1	85·7	94·3	102·9	111·4	120·0	128·6

The results of formula (13) are plotted on the accompanying diagram, from which it is clear that for all practical purposes the curve may be taken as a straight line, and the tabulated figures show that for most practical purposes Rankine's approximate formula is sufficiently accurate.

(2) *Dry saturated steam passes through an orifice 1 square inch in area from a vessel in which the pressure is 100 lbs. per square inch to a vessel at lower pressure. Plot a diagram showing the flow*

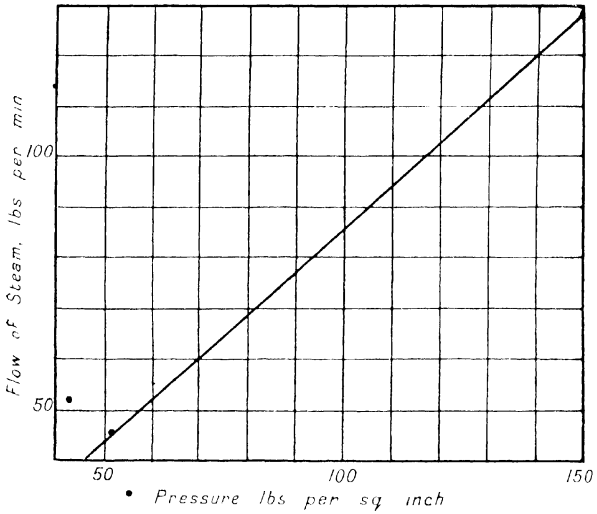


DIAGRAM SHOWING MAXIMUM FLOW OF STEAM THROUGH AN ORIFICE 1 SQ. INCH IN AREA FROM A CONTAINER AT VARIOUS PRESSURES.

of steam through the orifice when the pressure in the other vessel varies from 15 to 90 lbs. per square inch.

According to formula (8),

$$W = A \sqrt{\frac{2g\gamma}{(\gamma-1)} \cdot \frac{p_1}{V_1} \left\{ \left(\frac{p_2}{p_1}\right)^{\frac{2}{\gamma}} - \left(\frac{p_2}{p_1}\right)^{1+\frac{1}{\gamma}} \right\}},$$

$$i.e., \qquad W = A\sqrt{\frac{2g\gamma}{(\gamma-1)}\cdot\frac{p_1}{V_1}\left\{r_p^{\frac{2}{\gamma}} - r_p^{1+\frac{1}{\gamma}}\right\}}.$$

In our case $\gamma = 1{\cdot}135$, $p_1 = 100$ lbs. per square inch, $V_1 = 4{\cdot}40$ cubic feet per lb., $A = 1$ square inch, so that we have

$$W = 555\sqrt{r_p^{\frac{2}{1\cdot135}} - r_p^{\frac{2\cdot135}{1\cdot135}}} \text{ lbs. per minute.}$$

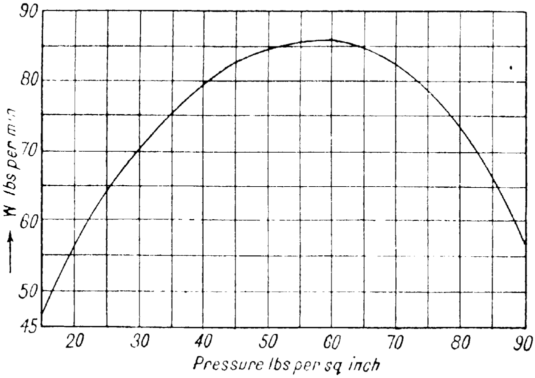


DIAGRAM SHOWING FLOW OF STEAM THROUGH AN ORIFICE 1 SQ. INCH IN AREA FROM A VESSEL AT 100 LBS. PER SQ. INCH TO A VESSEL AT DIFFERENT PRESSURES.

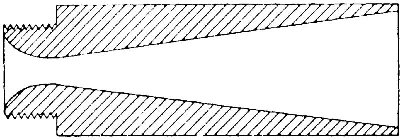
We can then tabulate as follows :—

p_2, . .	15	20	30	40	50	60	70	80	90
r_p, . .	·15	·20	·30	·40	·50	·60	·70	·80	·90
$r_p^{\frac{2}{1\cdot135}}$, . .	·035	·058	·120	·198	·295	·406	·533	·675	·830
$r_p^{\frac{2\cdot135}{1\cdot135}}$, . .	·028	·048	·104	·178	·272	·382	·511	·657	·820
$r_p^{\frac{2}{1\cdot135}} - r_p^{\frac{2\cdot135}{1\cdot135}}$,	·007	·010	·016	·020	·023	·024	·022	·018	·010
W, . .	46·8	56·1	70·1	79·5	84·7	85·8	82·5	73·8	56·4

On plotting these results we obtain the curve shown: from this curve it will be seen that the maximum flow occurs when the pressure in the second chamber is 60 lbs per square inch and is equal to 85·8 lbs per minute, thus agreeing with the result which we obtained in the previous example

Design of Steam Nozzles.—The velocity and volume of the steam both increase as the flow proceeds through a nozzle, and if we assume frictionless adiabatic flow to occur, it is easy to calculate the form of the nozzle so that the area of cross-section is equal to the volume passing per second divided by the velocity By this means eddying of the steam will be minimised.

Suppose that we have steam initially dry at a certain pressure p_1, then the dryness coefficient x at any other pressure p, assuming adiabatic flow, can be calculated from Mollier's diagram.



We can also calculate the velocity by the relation

$$\frac{v^2}{2g} = J(H_1 - H_2),$$

and, knowing the dryness coefficient, we can find the volume per lb of the steam

In practice, the size of the nozzle at the narrowest section or throat is determined from the number of lbs of steam which it is required to discharge per minute.

Then, by equation (13), we have

$$A = \frac{W}{18}\sqrt{\frac{V_1}{p_1}}$$

where A = area in square inches,

W = required discharge in lbs. per minute,

V_1 = volume of steam in cubic feet per lb,

p_1 = initial pressure of steam in lbs. per square inch.

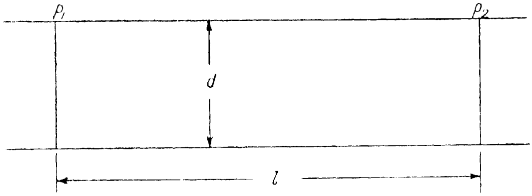
The area required at the outlet end at a given pressure can then be determined by the aid of the Mollier diagram in the manner previously referred to.

6

By continuing the expansion in a divergent nozzle, as used in the de Laval turbine, after the throat is passed, the amount of the discharge is not increased, but the steam is caused to acquire a greater exit velocity as it leaves the nozzle, because an increase is given to the range of pressure through which the expansion is effective for producing velocity.

Flow of Steam through Pipes. When a fluid flows through a pipe of constant cross section there is a loss in pressure due to friction along the sides of the pipe.

Experiments have shown the loss of pressure at comparatively low velocities is proportional to the square of the velocity of the fluid and to the length of the pipe, and is inversely proportional to the diameter of the pipe.



We may express this in symbols in the following manner :-

$$(p_1 - p_2) = \frac{f\, v^2\, l}{d}, \qquad \dots \quad (1)$$

where f = a constant depending on the roughness of the pipe,
v = velocity of fluid in the pipe,
d = diameter of pipe.

Now, if the quantity flowing per unit time is Q, and $A = \frac{\pi d^2}{4}$ is the area of the pipe, we have—

$$Q = A\, v = \frac{\pi\, d^2\, v}{4}.$$

$$\therefore \qquad v = \frac{4\, Q}{\pi\, d^2}.$$

$$\therefore \qquad (p_1 - p_2) = \frac{f\ 16\, .\, Q^2\, l}{\pi^2\, d^4\, .\, d},$$

or

$$Q = \frac{\pi}{4} \sqrt{\frac{d^5\, (p_1 - p_2)}{f\, .\, l}}. \qquad \dots \quad (2)$$

Experiments show that the constant f varies slightly with the diameter of the pipe, and is approximately proportional to $\left(1 + \frac{3.6}{d}\right)$.

The formula usually used in practice for the flow of steam is of the above type, and takes the form

$$W = 87 \sqrt{\frac{D\,(p_1 - p_2)\,d^5}{l\left(1 + \frac{3.6}{d}\right)}} \quad . \quad . \quad . \quad . \quad . \quad (3)$$

In this formula

- W = weight in lbs. per minute.
- D = weight in lbs. per cubic foot of steam.
- p_1, p_2 = pressures at beginning and end of pipe in lbs. per square inch.
- d = diameter of pipe in inches.
- l = length of pipe in feet.

According to the handbook, *Steam*, issued by Messrs. Babcock & Wilcox, Ltd., it has been found in practice that the most efficient speed through pipes is as follows:—

(*a*) For pipes up to 3 inches in diameter, 75 feet per second for saturated steam, and 100 feet per second for superheated steam.

(*b*) For pipes $3\frac{1}{2}$ to 9 inches in diameter, 90 feet per second for saturated steam and 120 feet per second for superheated steam.

(*c*) For pipes 10 inches in diameter and upwards, 90 feet per second for saturated steam and 140 feet per second for superheated steam.

In using the above figure, it should be borne in mind that the volume per lb. of superheated steam is greater than per lb. of saturated steam in the following ratios:—

° F. superheat,	100	150	200	250	300
Volume ratios,	1.15	1.23	1.30	1.36	1.44

The following table gives the weight of steam flowing for 1 lb. per square inch loss of pressure for various diameters of pipe and pressures of steam; for any other loss of pressure multiply by the square root of the pressure loss:—

Initial Pressure by Gauge, Pounds per Sq In.	Diameter of Pipe, in Inches Length of Each = 240 Diameters													
	¾	1	1½	2	2½	3	4	5	6	8	10	12	15	18
	Weight of Steam per Minute, in Pounds with One Pound Loss of Pressure.													
1	1·12	2·05	5·65	10·15	15·26	25·12	46·27	76·1	111·6	209·1	336·3	495·3	792	1,160
10	1·38	2·54	6·98	12·54	18·85	31·03	57·15	93·9	137·9	258·2	415·3	611·8	979	1,433
20	1·62	2·97	8·18	14·70	22·09	36·36	66·97	110·1	161·6	302·6	486·7	716·9	1,147	1,679
30	1·82	2·34	9·21	16·54	24·86	40·92	75·37	123·9	181·8	340·6	547·8	806·9	1,291	1,889
40	2·01	3·68	10·12	18·18	27·34	44·99	82·87	136·3	199·9	374·5	602·3	887·2	1,419	2,078
50	2·17	3·98	10·95	19·67	29·57	48·67	89·64	147·4	206·3	404·9	651·5	959·7	1,535	2,248
60	2·32	4·25	11·71	21·04	31·63	52·06	95·89	157·7	231·3	433·3	696·9	1026·5	1,642	2,404
70	2·46	4·51	12·42	22·32	33·55	55·22	101·71	167·3	245·4	459·6	739·3	1,088·9	1,742	2,550
80	2·59	4·75	13·09	23·52	35·36	58·19	107·18	176·3	258·6	484·3	778·9	1,147·4	1,836	2,687
90	2·71	4·96	13·66	24·55	36·91	60·74	111·88	183·9	269·9	505·5	811·5	1,197·8	1,916	2,805
100	2·84	5·20	14·32	25·73	38·71	63·66	117·25	192·8	282·9	529·8	852·2	1,255·3	2,008	2,940
120	3·06	5·61	15·44	27·75	41·71	68·64	126·43	207·9	305·1	571·3	918·9	1,353·6	2,166	3,170
150	3·37	6·16	16·97	30·49	45·83	75·42	138·91	228·4	335·2	627·7	1,009·6	1,487·2	2,379	3,483

Table giving flow of steam through pipes of various diameters.

Losses due to Elbows, Valves, etc. The loss of head due to getting up the velocity, to entering the pipe, and passing elbows and valves will reduce the flows given in the table. The resistance at entrance and due to a valve of usual type may be taken as that for a length equal to $\dfrac{114\,d}{1 + \dfrac{3\cdot6}{d}}$. For the sizes given in the above table these correspond to

Diameter of pipe, inches.	¾	1	1½	2	2½	3	4	5	6	8	10	12	15	18
Equivalent length in diameters.	20	25	34	41	47	52	60	66	71	79	84	88	92	95

The resistance at an elbow may be taken as equivalent to lengths equal to two-thirds of the above values.

Numerical Example. *Show that the horse-power transmitted in a steam pipe may be taken approximately as twice the flow of steam in pounds per minute.*

If a volume V cubic feet at pressure p lbs.-ft.$^{-2}$ passes in a minute, the work done per minute $= p\,V$ ft.-lbs.

$$\therefore \qquad \text{horse-power} = \frac{p\,V}{33{,}000}.$$

∴ if the weight passing per minute $=$ W, and the volume of steam per lb. $= V_1$, the volume $V = W V_1$, and we have, if p is the pressure per square inch,

$$\text{Horse-power} = \frac{144\,p\,V_1\,W}{33{,}000} = a\,W.$$

Now tabulate the following values from the steam tables:—

	V_1	a
50	8·42	1·84
100	4·40	1·92
150	3·01	1·97
200	2·29	2·00

The above rule, therefore, which states that a is approximately equal to 2, is proved to be correct.

Flow of Air through Pipes. When air flows through pipes or conduits a loss of pressure occurs owing to the friction between the air and the interior surfaces of the pipes. The resistance due to friction is generally considered to be proportional to the surface with which the air comes in contact—that is, for equal volumes passing, the loss due to friction varies directly as the length of the pipe and inversely to the diameter of the pipe. The loss also varies as the square of the velocity, and is best expressed by Weisbach's well-known formula:—

$$h = f\frac{l}{d} \times \frac{v^2}{2g}.$$

In which f = coefficient of resistance of friction determined by experiment,

l = length of pipe,

d = diameter of pipe,

v = velocity of the air,

g = acceleration due to gravity.

This is a variation of formula (1), p. 82.

The value of "f" naturally controls the result, and this coefficient depends both upon the material of which the pipe is constructed and the nature of the internal surface. In the case of a galvanised-iron pipe, which has been carefully made and erected with all internal laps extending in the direction of the air movement, the following formulæ, with constants in round numbers, have been deduced, and are employed by the Sturtevant Engineering Co., Ltd., from the general formula mentioned above:

$$p = \frac{l\,v^2}{25{,}000\,d} \qquad v = \sqrt{\frac{25{,}000\,d\,p}{l}}$$

$$l = \frac{25{,}000\,d\,p}{v^2} \qquad d = \frac{l\,v^2}{25{,}000\,p}$$

In all of which p = loss of pressure in ounces per square inch.

v = velocity in feet per second,

l = length of pipe in feet,

d = diameter of pipe in inches.

If we take the weight of 1 cubic foot of air in round numbers as 0·08 lb., and express the area of the pipe by A, the horse-power lost in friction in a pipe 100 feet long may be determined by the formula

$$\text{H.P.} = \frac{p\,A\,v}{8{,}800}.$$

By means of these formulæ the loss of pressure and horse-power lost in friction may be easily calculated for any size and length of pipe with air travelling at different velocities expressed in feet per minute. No allowance has been made for differences of temperatures between one end of the pipe and the other. In practice this is usually very small, and can be neglected.

For any other length of pipe other than 100 feet the loss of pressure and horse-power required will be directly proportional to the length. By working out a number of examples or by plotting curves from the above formulæ the desirability of making pipes of ample area will be emphasised. Suppose, for example, it is desired to move a given volume of air such that if passed through a 6-inch pipe it will give a velocity of 4,000 feet per minute, the horse-power lost in friction according to the formula would be 0·6316. If the same volume were passed through a 12-inch pipe, which, of course, has four times the area, the velocity would only require to be one-fourth as great or 1,000 feet per minute, and the loss in horse-power, according to the formula, would only be 0·0198, or one thirty-second of that expended in overcoming the resistance of a 6-inch pipe.

It is, of course, quite possible, and indeed desirable, to design the pipe for passing air so that it is the most satisfactory from every point of view, both as to first cost and as to the power required to move the air through the pipe. The best solution is obviously that which satisfies both conditions simultaneously. In many cases, especially when large volumes of air have to be passed through pipes of considerable length, very careful calculations should be made, in order to make sure that the design to be adopted is the most economical one possible.

It should be noted that the formulæ only give the horse-power which is absorbed due to the frictional resistance. For moving the air the power required for creating the necessary velocity must, of course, be added to that which will be lost in friction, in order to arrive at the total power necessary. The

dynamic or kinetic energy should, of course, be calculated from the ordinary formula—

$$H = \frac{D v^2}{2 g},$$

where H = the head required,

D = density of the air,

v = the velocity of flow,

g = acceleration due to gravity.

Lecture V.—Questions.

1. If steam at pressure (gauge) of 200 lbs. per square inch and counter-pressure 2·4 lbs. per square inch absolute is expanded adiabatically in a turbine nozzle, find the velocity of discharge of the steam and find the number of lbs. of steam per H P hour, if the steam leaves a perfect De Laval turbine with 34 per cent. of its initial velocity. *Ans.* $v = 3{,}780$ feet per second ; 10 lbs. of steam per hour.

2. Find an expression for the flow of steam through an orifice, neglecting friction.

3. Show how to determine, by aid of a $\tau\ \phi$ diagram, the dryness fraction of steam when expanded adiabatically. Dry steam, at an initial pressure of 100 lbs. per square inch absolute, is expanded adiabatically down to 20 lbs. absolute in a nozzle, which allows 1 lb. to pass through it per second. Determine the dryness of the steam and the sectional area of the nozzle at the lower pressure, using the data given in the accompanying table :—

Pressure in pounds per sq in	Temperature	Total Heat	Volume cubic feet	Entropy	
				Water	Steam
20	108·9° C. (228° F)	643·2 (C.H U.) (1,157·8° F) (B T.U.)	20 0	0·337	1 735
100	164 2° C. (327 6° F.)	662 0 (C.H.U.) (1,191 6° F) (B.T.U.)	4·44	0 475	1·609

Lecture V.—A.M.Inst.C.E. Questions.

1. A diverging nozzle is supplied with superheated steam at a given pressure, and expands the steam to a lower pressure, at which the quality is given. Show how to calculate the diameter at the throat and at exit for a given discharge and how to determine the velocity of the steam at exit.

2. Saturated steam at 100 lbs. atmospheric pressure ($t = 327·6°$ F.) is allowed to expand from one vessel to another, increasing its volume five-fold. Assume the vessels to have non-conducting walls and that the steam was initially dry, find the percentage of moisture at the end of the process. The volume of the water may be neglected. The specific volume of saturated steam is given by the equation:—

$$\text{Log}_e = 2·516 - 0·939 \log p.$$

$$\text{Latent heat of evaporation} = 1{,}092 - 0·7\,(t - 32).$$

$$\text{Lower temperature} = 225°\text{ F.}$$

LECTURE VI

MECHANICAL REFRIGERATION.

Reversed Heat Engines as Refrigerating Machines.—The ordinary heat engine, as we have already seen in studying the second law of thermodynamics, takes heat from a hot body, performs work, and rejects the remaining heat to a cold body.

If a heat engine be reversed it will absorb heat from the cold body and will transfer it to the hot body. According to one manner of expressing the second law of thermodynamics, heat cannot be transferred from a cold body to a hot body without the expenditure of work, and it is this work that the refrigerating machine has to perform.

A refrigerating machine, therefore, bears the same relation to an ordinary heat engine as a centrifugal pump does to a water turbine; for this reason refrigerating machines are often referred to as "heat pumps."

Coefficient of Performance.—The most economical refrigerating machine will clearly be the one which will extract the greatest number of heat units from the cold body for a given expenditure of mechanical work.

The quantity $\dfrac{\text{heat extracted}}{\text{work expended in heat units}}$ is called the *coefficient of performance* (P_c).

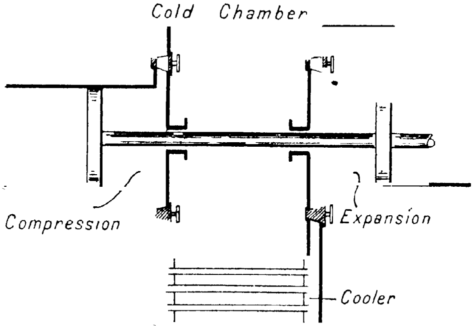
It follows from the second law of thermodynamics that if τ_h is the absolute temperature of the hot body and τ_c is that of the cold body, the maximum possible coefficient of performance will be—

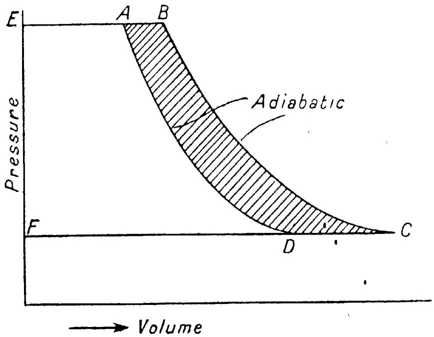
$$P_c = \frac{\tau_c}{\tau_h - \tau_c} \quad . \quad . \quad . \quad . \quad . \quad . \qquad (1)$$

For an engine, therefore, working to a given lower temperature

τ_c, the most efficient arrangement is that in which the range of temperature is small.

To cool a large quantity of a substance through a few degrees will require much less energy than to cool one-tenth of the amount through ten times as many degrees, although the amount of heat extracted is the same in both cases.





Air Refrigerating Machines—The Bell-Coleman Machine.—Refrigerating machines are of two main types—those employing air as a working substance and those employing vapours

In the Bell-Coleman machine, which was at one time used in many frozen meat steamships, the cycle employed is practically a reversed form of the Joule constant-pressure cycle.

The machine draws air from the cold chamber and compresses it; the heat resulting from the compression is removed by passing the compressed air through a cooler. The compressed air is then expanded and becomes chilled in the process, the cold air being then returned to the cold chamber. The amount of work which has to be expended in maintaining a certain temperature in the cold chamber depends, of course, upon the leakage of heat into it, and, therefore, upon the heat-insulating qualities of the walls of the chamber. Hollow walls with the cavities filled with powdered cork are commonly employed in practice.

The indicator diagram shows the cycle of operations, which can be summarised as follows:—

(1) Air is drawn in to the point C.

(2) The air is compressed adiabatically to B.

(3) The compressed air is driven into cooler and passes into the expander, the volume B E having contracted to E A, due to the cooling.

(4) The cooled compressed air is expanded adiabatically to D, and passes into the cold chamber.

It will be clear from the analysis of the constant-pressure cycle given on p. 62, the maximum possible coefficient of performance of this engine is given by—

$$P_0 = \frac{\tau_D}{\tau_A - \tau_D} = \frac{\tau_C}{\tau_B - \tau_C} = \frac{1}{r^{\frac{\gamma-1}{\gamma}} - 1}.$$

Numerical Example.—*Find the maximum possible coefficient of performance of a Bell-Coleman refrigerating engine which compresses from one to four atmospheres, the temperature after compression being reduced to 60° F. How much ice from and at 32° F. can be produced per I.H.P. hour?*

Referring to the diagram on p. 92, we have—

$$\tau_A = 461 + 60 = 521.$$

$$\frac{\tau_D}{\tau_A} = \left(\frac{p_D}{p_A}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{4}\right)^{\frac{\cdot 4}{1\cdot 4}}.$$

$$\tau_D = \frac{\tau_A}{4^{\frac{\cdot 4}{1\cdot 4}}} = \frac{521}{1\cdot 486} = 350° \text{ Fah.} = -111° \text{ F.}$$

$$\therefore \quad P_C = \frac{\tau_D}{\tau_A - \tau_D} = \frac{350}{521 - 350} = \frac{350}{171} = \mathbf{2{\cdot}05}.$$

$$\text{I.H.P.-hour} = \frac{33{,}000 \times 60}{778} \text{ B.Th U.}$$

$$= 2{,}550 \text{ B Th U approx.}$$

The heat to be deducted per lb. from and at 32° F = latent heat of ice = 142 B Th U

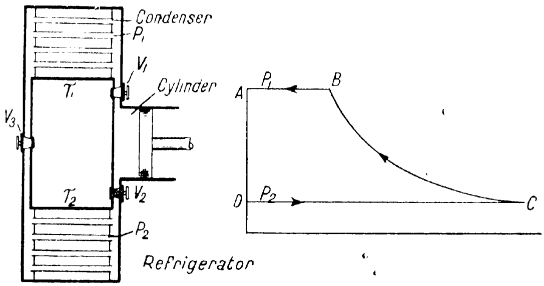
Now, heat extracted = work expended × coefficient of performance.

∴ If x lbs are converted into ice

$$142\,x = 2{,}550 \times 2{\cdot}05.$$

$$x = \frac{2{,}550 \times 2{\cdot}05}{142} = \mathbf{36{\cdot}8} \text{ lbs.}$$

Vapour Refrigerating Machines.—In most modern refrigerating machines the working substance, instead of being air, consists of a liquid of low boiling point such as carbonic acid or carbon dioxide (CO_2), ammonia (NH_3), and sulphurous acid or sulphur dioxide (SO_2), the thermal properties of which are given on pp. 10-12.



These engines,* therefore, correspond to steam engines, and they usually follow a cycle corresponding to the Rankine-Clausius cycle run backwards.

At this stage we will remind the student that there is thermodynamically a distinction between a "reversible" cycle and one

* See also diagrammatic view on p. 99.

which can be worked in a reverse direction. The Rankine-Clausius cycle, for instance, is not "reversible" in the sense defined on p. 12, but it can be run in the reverse direction; for a cycle to be "reversible," all the heat must be taken in at the upper temperature limit, and all the heat must be rejected at the lower temperature limit.

The essential portion of the machine consists in the cylinder or compressor, the condenser, and the refrigerator or evaporator.

The first portion of the cycle is represented by D C on the diagram, and consists in the outward movement of the piston thus drawing vapour at pressure p_2 and temperature τ_2 into the cylinder. During this phase the valve V_1 is closed and the valve V_2 is open.

The piston then moves in and compresses the vapour approximately adiabatically until its pressure is p_1, both valves V_1 and V_2 being closed; this operation is represented by C B on the p V diagram. The valve V_1 is then opened and the vapour passes into the condenser; this condenser is usually kept cool by water surrounding a coil of pipes constituting the condenser, and the heat generated by the compression is withdrawn by the water, so that the vapour condenses under the pressure to which it is subjected; this phase is represented by B A on the p V diagram.

To complete the cycle, the valve V_3 is opened, and an amount of liquid equal to that which has been added by condensation passes down into the refrigerator, the pressure falling from p_1 to p_2; this stage is represented by A D on the p V diagram.

Ideal Coefficient of Performance of Vapour Refrigerating Machines.—We can deduce the ideal coefficient of performance for a refrigerating machine working according to the cycle previously described by the following consideration of the temperature-entropy diagrams.

Three cases should be considered:—

(1) Vapour wet at end of compression.

(2) Vapour saturated at end of compression.

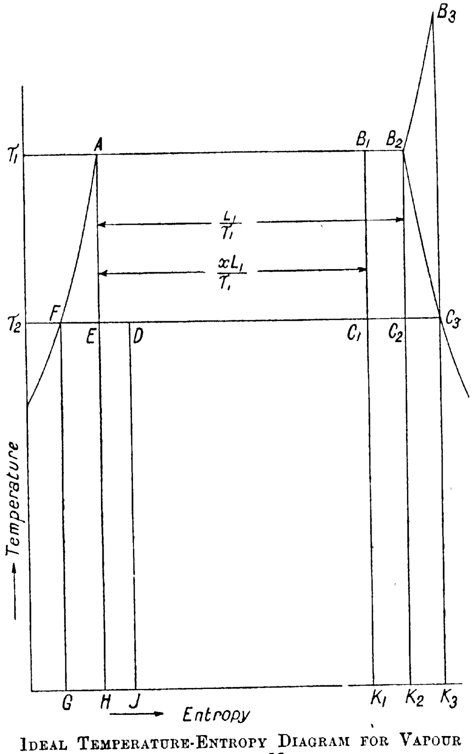
(3) Vapour saturated at beginning and superheated at end of compression.

Corresponding to these three cases we have the compression lines $C_1 B_1$, $C_2 B_2$, and $C_3 B_3$ on the $\tau \varphi$ diagram.

The work done in the compression cylinder per cycle in the three cases is represented by the areas A F C_1 B_1, A F C_2 B_2, and

A F C_3 B_2, and the total heat taken from the refrigerator per cycle is represented by the areas F G K_1 C_1, F G K_2 C_2, and F G K_3 C_3.

But as the liquid at temperature τ_1 passes through the throttle valve into the refrigerator at temperature τ_2, it will give back



IDEAL TEMPERATURE-ENTROPY DIAGRAM FOR VAPOUR REFRIGERATING MACHINE.

thereto an amount of heat equal to s $(\tau_1 - \tau_2)$, where s is the specific heat. This heat is represented by the area F G J D, so that the net area withdrawn per stroke in the three cases

will be represented by the areas $D\,J\,K_1\,C_1$, $D\,J\,K_2\,C_2$, and $D\,J\,K_3\,C_3$.

Case I.—Vapour Wet at End of Compression.—In this case the closing line of the $\tau\,\varphi$ diagram is $C_1\,B_1$, the dryness coefficient x at the close of the compression being represented by the fraction $x = \frac{A\,B_1}{A\,B_2}$.

We then have—

$$\text{Coefficient of performance} = P_C = \frac{\text{heat extracted}}{\text{work done}}$$

$$= \frac{\text{area } D\,J\,K_1\,C_1}{\text{area } A\,F\,C_1\,B_1}.$$

$$\text{Now, area } D\,J\,K_1\,C_1 = \text{area } F\,G\,K_1\,C_1 - \text{area } F\,G\,J\,D$$

$$= \text{area } F\,G\,H\,E + \text{area } E\,H\,K_1\,C_1 - \text{area } F\,G\,J\,D$$

$$= F\,G\,(F\,E + E\,C_1) - s\,(\tau_1 - \tau_2)$$

$$= \tau_2 \left(s \log_e \frac{\tau_1}{\tau_2} + \frac{x\,L_1}{\tau_1} \right) - s\,(\tau_1 - \tau_2) \quad (1)$$

$$\text{And, area } A\,F\,C_1\,B_1 = \text{area } A\,F\,G\,H + \text{area } A\,H\,K_1\,B_1 - \text{area } F\,G\,K_1\,C_1$$

$$= s\,(\tau_1 - \tau_2) + \frac{x\,L_1}{\tau_1}\,.\,\tau_1 - \tau_2 \left(s \log_e \frac{\tau_1}{\tau_2} + \frac{x\,L_1}{\tau_1} \right)$$

$$= \left(s + \frac{x\,L_1}{\tau_1} \right) (\tau_1 - \tau_2) - s\,\tau_2 \log_e \frac{\tau_1}{\tau_2}. \quad (2)$$

$$P_C = \frac{\tau_2 \left(s \log_e \frac{\tau_1}{\tau_2} + \frac{x\,L_1}{\tau_1} \right) - s\,(\tau_1 - \tau_2)}{\left(s + \frac{x\,L_1}{\tau_1} \right) (\tau_1 - \tau_2) - s\,\tau_2 \log_e \frac{\tau_1}{\tau_2}}. \quad (3)$$

Case II.—Vapour Saturated at End of Compression.—In this case we have—

$$\text{Coefficient of performance} = \mathrm{P_C} = \frac{\text{area D J K}_2\,\text{C}_2}{\text{area A F C}_2\,\text{B}_2}$$

$$= \frac{\tau_2\left(s\log_e\dfrac{\tau_1}{\tau_2} + \dfrac{\mathrm{L}_1}{\tau_1}\right) - s\,(\tau_1 - \tau_2)}{\left(s + \dfrac{\mathrm{L}_1}{\tau_1}\right)(\tau_1 - \tau_2) - s\,\tau_2\log_e\dfrac{\tau_1}{\tau_2}}. \quad . \quad . \quad (4)$$

This result is obtained by putting $x = 1$ in equation (3).

Case III.—Vapour Saturated at Beginning and Superheated at End of Compression. In this case we have—

$$\text{Coefficient of performance} = \mathrm{P_C} = \frac{\text{area D J K}_3\,\text{C}_3}{\text{area A F C}_3\,\text{B}_3\,\text{B}_2}.$$

Now, if s' is the specific heat of the superheated vapour and t is the amount of the superheat, then we shall have—

$$\mathrm{K_2\,K_3} = \mathrm{C_2\,C_3} = s^1\log_e\frac{\tau_1 + t}{\tau_1}.$$

$$\text{Area } \mathrm{B_2\,K_2\,K_3\,B_3} = s^1\,t.$$

$$\text{Area } \mathrm{C_2\,K_2\,K_3\,C_3} = \tau_2\,s^1\log_e\frac{\tau_1 + t}{\tau_1}.$$

$$\text{Area } \mathrm{B_2\,C_2\,B_3\,C_3} = s^1\,t - \tau_2\,s^1\log_e\frac{\tau_1 + t}{\tau_1}.$$

$\therefore$ Area D J $\mathrm{K_3\,C_3}$

$$= \text{area D J } \mathrm{K_2\,C_2} + \text{area } \mathrm{C_2\,K_2\,K_3\,C_3}$$

$$= \tau_2\left(s\log_e\frac{\tau_1}{\tau_2} + \frac{\mathrm{L}_1}{\tau_1}\right) - s\,(\tau_1 - \tau_2) + \tau_2\,s^1\log_e\frac{\tau_1 + t}{\tau_1}.$$

$$= \tau_2\left(s\log_e\frac{\tau_1}{\tau_2} + s^1\log_e\frac{\tau_1 + t}{\tau}\right) - s\,(\tau_1 - \tau_2). \quad . \quad . \quad (5)$$

Area A F $\mathrm{C_3\,B_3\,B_2}$

$$= \text{area A F } \mathrm{C_2\,B_2} + \text{area } \mathrm{B_2\,C_2\,C_3\,B_3}$$

$$= \left(s + \frac{\mathrm{L}_1}{\tau_1}\right)(\tau_1 - \tau_2) - s\,\tau_2\log_e\frac{\tau_1}{\tau_2} + s^1\,t - \tau_2\,s^1\log_e\frac{\tau_1 - t}{\tau_1}$$

$$= \left(s + \frac{\mathrm{L}_1}{\tau_1}\right)(\tau_1 - \tau_2) + s^1\,t$$

$$- \tau_2\left(s\log_e\frac{\tau_1}{\tau_2} + s^1\log_e\frac{\tau_1 + t}{\tau_1}\right). \quad . \quad (6)$$

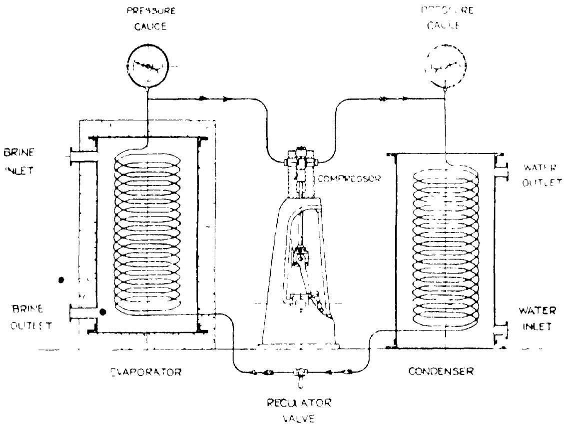
$\therefore$ Coefficient of performance $= P_c$

$$= \frac{\tau_2\left(s\log_e\frac{\tau_1}{\tau_2} + s^1\log_e\frac{\tau_1+t}{\tau}\right) - s(\tau_1 - \tau_2)}{\left(s + \frac{L_1}{\tau_1}\right)(\tau_1 - \tau_2) + s^1 t - \tau_2\left(s\log_e\frac{\tau_1}{\tau_2} + s^1\log_e\frac{\tau_1+t}{\tau_1}\right)} \qquad (7)$$

Typical Description of Vapour Compression Machines adopted in Practice.—We are indebted to Messrs J & E Hall, Ltd, Dartford, Kent, for the following particulars of typical machines.

As a general rule carbon dioxide machines are recommended for use on ships and ammonia machines for land use.

The following diagrammatic illustration shows the essential parts of the plant.



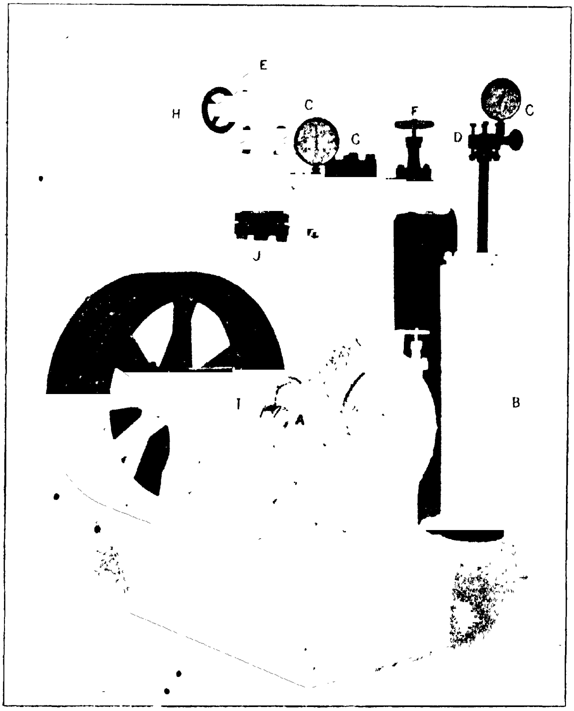
The cycle of operations is as follows. On the suction stroke of the compressor piston a charge of gas is drawn from the evaporator and on the return stroke the gas is compressed and discharged into the condenser coil at a pressure sufficient to cause liquefaction, this pressure depending directly on the temperature of the cooling water. The gas enters the condenser coils at a high temperature and during its passage through the coils is cooled to a temperature within a few degrees of the cooling water circulating over the surface of the coils, with the result

that the gas is liquefied, the latent heat of liquefaction passing into the surrounding water. The liquid CO_2 or NH_3 then passes from the lower terminals of the condenser coils through the regulating valve to the evaporator coils, where evaporation takes place at a temperature sufficiently below that of the medium to be cooled to allow of a reasonably rapid interchange of heat. The evaporator coils thus perform the reverse function of the condenser coils, the heat required to evaporate the liquid CO_2 or NH_3 being absorbed from the surrounding brine solution or other medium, which is consequently lowered in temperature. The outlet terminals of the evaporator coils are connected to the suction of the compressor, the CO_2 or NH_3 passing in the form of a gas from the evaporator to the compressor and thus completing the cycle.

Details of Ammonia Compressor.- The enclosed type machine illustrated herewith is employed for small installations.

The cylinder and crank-case are cast in one piece, the cylinder being fitted with a removable liner The valves are of the mushroom type, and are arranged in the top cover, so that they can be readily withdrawn and replaced without disturbing the pipe connections. The piston is of the ordinary trunk type, and is fitted with four cast-iron rings. The crank is of the overhung type, and the crank-shaft passes through a gland, which must, of course, be maintained gas-tight. The ammonia pressure in the crank-case under normal conditions of working is comparatively low, corresponding approximately to the pressure in the evaporator coils, which again corresponds to the temperature of expansion. For ordinary cases, where the machine is maintaining a temperature of, say, 35° in a small cold chamber, the pressure would amount to about 30 lbs. per square inch. Thus, with a gland of the rotary type no difficulty is found in preventing leakage of ammonia at this point. Lubrication of the machine is of the splash type, the crank-case being fitted with an oil-inspection glass, to enable the oil level to be maintained at the correct height. The machine is fitted with an oil separator, where any oil passing to the top of the piston and through the discharge valve is trapped before passing into the condenser coils. A connection between the oil separator and crank-case enables the oil to be discharged periodically from this vessel back into the crank-case. The crank-shaft is carried in an outer bearing, and is fitted with fast and loose pulleys for belt drive.

The type of condenser most commonly used with these small machines is the double pipe condenser, which is somewhat more economical in water consumption than a condenser of the submerged type owing to the counter-current flow principle, which

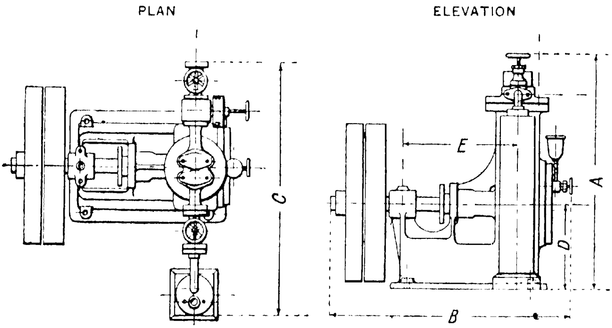


NH$_3$ Refrigerating Machine

A, Oil Level Indicator.
B, Oil Separator.
C, C, Gauges.
D, Discharge to Condenser.
E, Inlet from Evaporator
F, Stop Valve (Delivery).
G, Compressor Valve Covers.
H, Stop Valve (Suction).
I, Gland
J, Suction Strainer.

is made use of, thus enabling a greater amount of heat to be discharged into a given quantity of cooling water while still maintaining the condensed liquid at a temperature only slightly above the temperature of the condensing water, than is the case with the submerged condenser.

The condenser consists of an inner coil of wrought-iron tube electrically welded into a continuous length and surrounded by an outer pipe casing, the condensation of the ammonia taking place within the inner coil, and the annular space between the pipe forming a passage for the condensing water. The condenser to be fitted with wrought-iron bar supports suitable for fixing either to the wall or on the floor.



HALL'S AMMONIA REFRIGERATING MACHINES.

Vertical Single-Acting Enclosed Type.

Where the water consumption is limited, a condenser of the atmospheric or evaporative type is supplied where the same water is continually recirculated over the coils, and the consumption of water is merely that lost by evaporation.

The form of the evaporator depends on the purpose for which the plant is to be used, and may consist of direct expansion coils arranged in a cold chamber or coils fitted in a tank surrounded by a solution of calcium chloride, which may be pumped through piping fitted in the cold chamber.

Loading Dimensions and Data.—The following tabulated data of the machine previously described will be of interest to students.

Size of Machine.	Overall Dimensions A	B	C	D	E
1	3′ 2¾″	3′ 2″	3′ 2″	1′ 1″	1′ 6⅝″
2	3′ 11½″	3′ 6½″	3′ 3¾″	1′ 3″	1′ 7½″
3	3′ 11½″	3′ 6½″	3′ 3¾″	1′ 3″	1′ 7½″
4	3′ 11½″	3′ 6½″	3′ 3¾″	1′ 3″	1′ 7½″
5	4′ 2½″	4′ 0½″	3′ 8½″	1′ 4½″	1′ 10¼″
6	4′ 2½″	4′ 0½″	3′ 8½″	1′ 4½″	1′ 10¼″
7	5′ 2½″	5′ 4¼″	4′ 0¾″	1′ 9″	2′ 9″
8	5′ 2½″	5′ 4¼″	4′ 0¾″	1′ 9″	2′ 9″

Size of Machine	Compressor Details: Diam of Piston.	Stroke	R P M	Suction and Delivery Pipe	Pulleys: Diam	Face
1	3½″	4″	200	¾″	2′ 0″	3″
2	4″	6″	140	1″	2′ 3″	4½″
3	4″	6″	170	1″	2′ 3″	4½″
4	4″	6″	200	1″	2′ 3″	4½″
5	5″	6″	170	1¼″	2′ 6″	5½″
6	5″	6″	200	1¼″	2′ 6″	5½″
7	6″	8″	140	1½″	3′ 3″	6½″
8	6″	8″	170	1½″	3′ 3″	6½″

Size of Machine.	Rev Power Tons per Day	Ice Making Capacity per 24 Hours: Water 55° F	Water, 65° F	Water 75° F	Water, 85° F	Water, 95° F
		Cwts.	Cwts	Cwts	Cwts	Cwts
1	1·0	8·0	7·3	6·8	6·3	5·6
2	1·5	12·0	11·0	10·4	9·5	8·5
3	1·75	15·0	13·7	13·0	11·8	10·5
4	2·0	18·0	16·5	15·5	14·2	12·7
5	2·5	25·0	23 0	21·7	19·7	17·5
6	3·0	30 0	27·5	26 0	23 6	21·0
7	4·0	40·0	36·0	32 0	30·0	24·0
8	5·0	44·0	40·0	38 0	34·0	31·0

Size of Machine.	Approximate Internal Capacity of Chamber in Cubic Feet		
	Temperate Climate and 6 in. Insulation	Sub-Tropical Climate and 8-in Insulation	Tropical Climate and 10 in Insulation
1	600	500	400
2	1,000	800	700
3	1,500	1,300	1,000
4	2,000	1,800	1,500
5	3,000	2,500	2,000
6	4,500	3,750	3,000
7	7,000	6,000	5,000
8	10,000	8,000	7,000

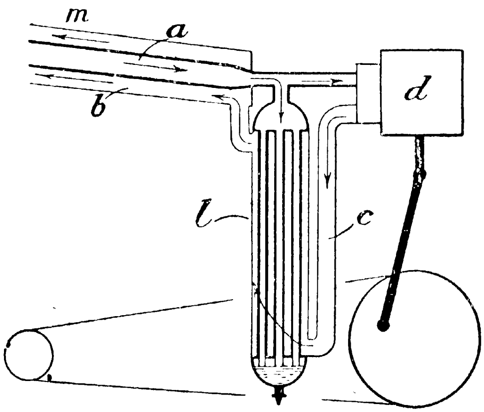
Size of Machine	Power Required under Cold Storage and Ice Making Conditions Compressor B H P				
	55° F	65° F	75° F.	85° F	95° F
1	2·0	2 05	2 1	2 15	2 2
2	2 8	2·9	3 0	3 1	3·2
3	3·4	3 55	3·7	3·85	4 0
4	4 2	4 4	4·55	4·7	4·85
5	5·0	5·2	5·4	5·6	5·8
6	5·6	5 8	6·1	6·4	6 6
7	6 5	7 5	8 2	8 5	9·0
8	7 4	8·2	9 5	10 2	11 0

The Production of Very Low Temperatures. An interesting application of the principles of refrigeration occurs in the production of very low temperatures for the production of liquid air. The modern method for producing oxygen in practice consists in liquefying air and in separating out the nitrogen, argon, etc., by the process of fractional distillation. These low temperatures are obtained by the expansion of compressed air. The expansion may be effected against the piston of a machine working under load, when the work done by the air in overcoming the resistance to increase of volume involves an abstraction of heat from the gas, with a consequent fall in temperature; or the air may be allowed to expand from the initial high pressure

to the final low pressure by passage through a restricted orifice or valve, without performing external work. The first method is usually attributed to Claude, and the second to Linde.

Claude Process. The accompanying figure shows in diagrammatic form the principle involved in the Claude process.

The compressed air at a pressure of 40 atmospheres passes through the inner tube *a* of the heat exchanger *m* to the expansion machine *d*. The expanded and cooled air then passes upwards round the outside of the tubes of the liquefier *l*. These tubes are supplied with the compressed air at 40 atmospheres pressure from the tube *a*. This compressed air is thus progressively



cooled by the expanding gases circulating upwards until the temperature of liquefaction at that temperature—about −140° C.—is reached.

Liquefaction then commences in the tubes, the liquid collecting in the bottom of the liquefier, from which it can be run off by means of a cock. The expanded gas passes round the tubes of the liquefier, and thus attains the temperature of liquefaction of the compressed gas; it then passes into the outer tube *b* of the exchanger, and thus cools the incoming compressed gas, which, therefore, reaches the expanding machine at this temperature.

The cold air issuing from the expansion engine at a temperature

not far removed from its point of liquefaction, and at a pressure of about four atmospheres, passes to the outer compartment A of the bottom liquid-collecting vessel of the column. It then ascends the vertical nest of tubes B leading from the top of this compartment, and which are immersed in baths of liquid oxygen C and D. Here it undergoes liquefaction, not as a whole, but progressively. The first drops of liquid formed at the bottoms of the tubes are considerably richer in oxygen than air, for the proportion of the less volatile oxygen in any oxygen-nitrogen liquid is greater than its proportion in the gas in equilibrium with the liquid. Higher up the tubes the liquid condensed out becomes, progressively poorer in oxygen. But this liquid falling back comes into contact with the ascending gaseous mixture which is too rich in oxygen for equilibrium. Nitrogen is, therefore, evaporated from these falling drops, oxygen being condensed in corresponding amount. The height of the tubes is so arranged that the gas passing out at their upper ends is very rich in nitrogen, which travels down a similar nest of tubes leading to the inner compartment E of the bottom vessel, which it reaches partly in the liquid form. The incoming air is thus separated into two portions on liquefaction, one containing in practice about 40 per cent. of oxygen, and the other only 4 to 5 per cent. In the extreme centre of the tubes is a small bundle F, through which a quantity of gas from the inner compartment can be withdrawn; after this third purification, a relatively small nitrogen product can be obtained of 99·5 to 100 per cent. purity. The pressure of 4 atmospheres is necessary in the tubes to raise the liquefying points of the oxygen-nitrogen mixtures above $-183°$ C., the temperature of the liquid oxygen bath, in order that liquefaction may take place and that the atent heat may pass to the liquid oxygen, which is evaporated

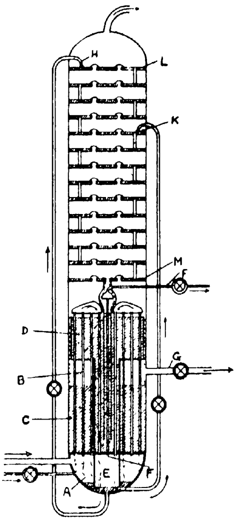


Diagram of Claude's Oxygen Column.

to a corresponding extent. Part of this evaporated oxygen is led away by G through the interchanger to a gasholder, to furnish the oxygen product, and the rest ascends the rectifying column, bubbling through the liquid which is passing down over the plates. This liquid has its origin in the rich and poor liquids just considered, which are conveyed into the rectification column from the bottom vessel, the poor liquid at the very summit H, and the rich some distance down, at K. Between the top plate, L, containing almost pure nitrogen at a temperature near $-195°$ C., and the bottom plate, M, containing almost pure liquid oxygen at a temperature of about $-183°$ C., there exists a temperature difference of some 12°. As the gaseous oxygen rises into the colder regions of the column, it undergoes condensation, its heat of liquefaction evaporating an equivalent quantity of the more volatile nitrogen. Thus, the descending liquid is becoming progressively richer in oxygen, and the ascending gases are becoming progressively impoverished. The rich liquid containing about 40 per cent. of oxygen is in equilibrium with a gas mixture having an oxygen content of about 15 per cent. It is, therefore, introduced into the column at the point K, where the gases have this composition, and in falling can scrub the rising gases to this extent. The poor liquid with 4 to 5 per cent. of oxygen, introduced at the top, H, of the column, can scrub the rising gases theoretically down to 98 per cent. nitrogen. In practice, however, this poor liquid is not produced in sufficient quantity. In order to obtain a high quality of oxygen, the argon impurities being difficult to remove owing to the similarity of the volatilities of liquid argon and oxygen, the nitrogen leaves the apparatus still containing 6 to 7 per cent. of this gas. By taking more oxygen away to the gasholder, of lower purity, and so diminishing the quantity that has to ascend the column, the final scrubbing becomes more effective, and 97 to 98 per cent. nitrogen can be obtained.

For further particulars of this type of apparatus the student should consult a paper by Mr. C. R. Houseman, of the British Oxygen Company, Limited, entitled "The Evaporation of Air Constituents," given in *Industrial Gases* for March, 1920.

Lecture VI.—Questions.

1. Describe the action of the Bell-Coleman refrigerating machine, and find an expression for its coefficient of performance.

2. Find the H.P. theoretically necessary in an air-compression refrigerating machine to abstract Q thermal units per minute, where t_1 = temperature of air drawn into compressor, t_2 = temperature of air forced by the compressor into the cooler, t_3 = temperature of air supplied to the expanding cylinder, and t_4 is the temperature of the cold air leaving the expanding cylinder.

3. Describe by aid of a diagram the principle of action of a refrigerating machine of the open-cycle air type, and obtain an expression for its efficiency, assuming adiabatic compression and expansion. In a machine of this type circulating 1,500 lbs. of air per hour, the air is drawn from a cold chamber at a temperature of 10° C. (50° F.), and compressed adiabatically to 67 lbs. absolute. It is afterwards cooled at this pressure to 25° C. (77° F.), the temperature of the condenser, and then expanded adiabatically to atmospheric pressure and returned to the cold chamber. Find the number of units of heat extracted per hour from the cold chamber and the heat rejected. If the indicated horse-power of the compressor is 25·0, find the coefficient of performance of the machine. The specific heat of air may be taken as 0·241, and the law of expansion and compression as $p v^{1.4}$ = constant.

4. The following are approximate expressions for the entropy of ammonia liquid and dry saturated vapour:—Liquid, 0·00184 $(t - 32)$; vapour, 1·158 − 0·00192 $(t - 32)$, t being the temperature on the Fahrenheit scale. Obtain corresponding expressions of the form $a + b\,t$, t, being the temperature on the Centigrade scale. Draw the θ-φ chart between temperatures of 14° F. and 77° F. (−10° C. and 25° C.). Find the coefficient of performance of a refrigerator working on a reversed Rankine cycle between these limits, the vapour being 5 per cent. wet at the end of compression. If the actual performance is 0·6 of the amount in the above ideal case, calculate the pounds of ice produced per horse-power hour from water at the freezing point. Latent heat of ice, 144 B.Th.U. (80 C.H.U.)

5. Find an expression for the coefficient of performance of a refrigerating machine working on the Bell-Coleman cycle. Explain the reasons of the smallness of the coefficient in practice when compared with an ammonia refrigerator.

6. Describe the action of some type of refrigerating machine working on the compression system. What do you understand by "wet compression"? In choosing a suitable fluid for use in the compressor, state the effect of the following on the general efficiency of the plant:—Latent heat, specific heat, specific volume, relation between pressure and temperature.

LECTURE VI.—A.M.INST.C.E. QUESTIONS

1. Sketch the temperature-entropy diagram for ammonia, and show on it the refrigerating cycle for wet compression. How is the coefficient of performance determined?

2. Draw p-v and θ-ϕ diagrams for a refrigerating machine, using air, which has admission and exhaust at constant pressures, and adiabatic compression and expansion. If the temperatures at beginning and end of compression are 20° F. and 390° F., and the temperature at the beginning of expansion is 100° F., find the coefficient of performance.

3. Describe and give hand sketches of an ammonia refrigerating machine. Describe the gland arrangements that have to be adopted to prevent leakage of the ammonia.

4. Explain how you would test a refrigerating plant using ammonia as the working fluid, so as to determine the coefficient of performance.

5. Describe the cycle of operations of either a carbon dioxide or an ammonia refrigerating machine, and state the special advantages of each type.

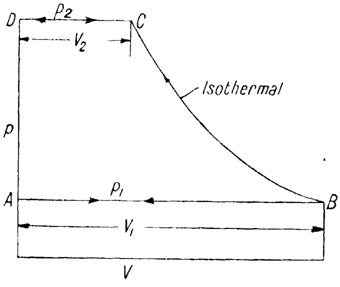
6. In an ice-making plant of either (*a*) the ammonia or (*b*) CO_2 type explain the method of working, and describe the operations necessary to start up and regulate the machine from the commencement until ice began to form.

7. Give briefly the advantages and disadvantages of ammonia and carbon dioxide respectively as working fluids in refrigerators. Make a diagrammatic sketch, illustrating the construction of a plant using either carbon dioxide or ammonia

LECTURE VII.

COMPRESSED AIR.

Isothermal and Adiabatic Compression and Expansion. Power is transmitted by compressed air for a large number of industrial purposes, particularly for driving hand tools, such as drills, chipping chisels, and riveters. It has been used on a large scale



in Paris for supplying power from a central station, but it is, on the whole, expensive, and is used mainly on account of its convenience for delivering small powers, or in places where the cold exhaust can be used for refrigeration. Professor Unwin estimates that when used on a large scale 44 to 51 per cent. of the indicated steam power of the engines driving the compressors may be realised on the main shaft of the compressed air engine, but in small motors and rock drills a much smaller efficiency is usually obtained.

If the air compressor worked so slowly that the compression were isothermal, the temperature of the compressed air would

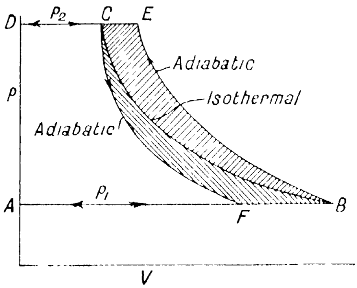
be the same as the intake temperature (in practice the air always gets hot in compression), and the volume would, therefore, not diminish by cooling on its way to the compressed air engine.

The indicator diagram of the air compressor would be as shown in the above diagram.

Air is drawn into the cylinder at temperature p_1 until the volume is represented by the point B, it is then compressed isothermally along the line B C until its pressure is p_2, and is driven out at constant pressure along the line C D

If the compressed air engine also works so slowly that the expansion is isothermal (in practice the air always cools in expansion), then the engine would trace out the diagram in the reverse direction, and the efficiency would be 100 per cent. if there were no loss of pressure due to friction in pipes, etc.

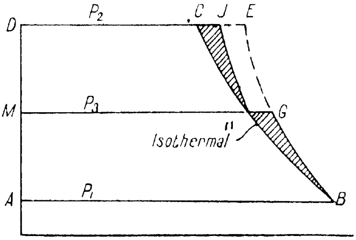
Now, consider the case in which the compression and expansion are adiabatic. The compression curve will be steeper than the



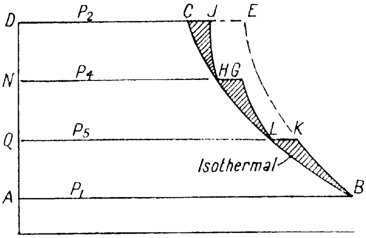
isothermal, and may be represented by B E; the heated air will contract on cooling, so that by the time it reaches the compressed air engine the volume will have become D C and the expansion will follow the curve C F. It is, therefore, clear that the work represented by the shaded area C F B E will be lost, of which B E C may be regarded as lost in the compressor and C F B in the engine.

The Advantages of Multi-Stage Compression.—The following diagrams show how this loss may be reduced by the employment of multi-stage compression, a corresponding advantage is to be obtained by multi-stage expansion.

Take, first, the case of two-stage compression; the air is first compressed to an intermediate pressure p_3 along the adiabatic B G, and passes into a receiver in which the temperature falls until the volume is M H; the air then passes into a second cylinder, in which it is compressed to the pressure p_2.



2 Stage Compression



3 Stage Compression

The loss of work in the compression is represented by the sum of the areas B G H and H J G, and this is clearly less than the area B E C, which would be lost in single-stage compression.

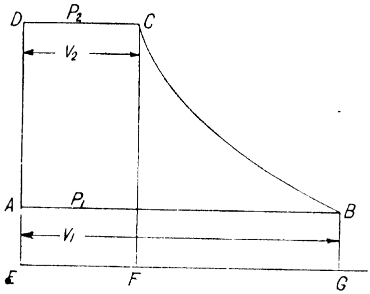
Water Injection to Assist Cooling.—In practice it is found to be useful to inject water in the form of a spray in air compressors to keep down the temperature during compression. This has the effect of making the compression curve follow a line between the adiabatic and isothermal lines, and some authorities assume

the compression in this case to follow the law $p\,V^{1\cdot 2}$ = constant instead of $p\,V^{1\cdot 41}$ = constant (adiabatic compression). Other authorities prefer to make their calculations upon the basis of $p\,V^{1\cdot 41}$ = constant, even when water injection is employed to make up for incidental losses which are found to occur in practice.

One disadvantage attendant upon the use of water in air compressors arises in the formation of snow in the compressed-air engines when the pressure drop is considerable; this snow tends to clog the exhaust valves. The difficulty can be overcome in part by making the valves and passages of large size; in some cases the formation of snow has been prevented by warming the compressed air supply pipe

Horse-Power required for Compressing Air.—(1) *Single-stage Compression.*—The work done in compression = A B C D

$$= \text{B C F G} + \text{F C D E} - \text{G B A E}.$$



By equation (6), p. 6, B C F G $= \dfrac{p_2\,V_2 - p_1\,V_1}{(n-1)}$;

$\therefore$ Work done or energy E

$$= \frac{p_2\,V_2 - p_1\,V_1}{(n-1)} + p_2\,V_2 - p_1\,V_1,$$

i.e.,
$$E = \frac{n}{n-1}\,(p_2\,V_2 - p_1\,V_1). \qquad (1)$$

Now, $p_2\,V_2^n = p_1\,V_1^n$,

$$\therefore \qquad V_2 = \left(\frac{p_1}{p_2}\right)^{\frac{1}{n}} . V_1,$$

$$\therefore\ p_2 V_2 - p_1 V_1 = V_1 \left\{ p_2 \left(\frac{p_1}{p_2}\right)^{\frac{1}{n}} - p_1 \right\}.$$

$$= V_1 \{ p_2^{1-\frac{1}{n}} \cdot p_1 \cdot p_1^{-1+\frac{1}{n}} - p_1 \}$$

$$= p_1 V_1 \{ p_2^{\left(1-\frac{1}{n}\right)} \cdot p_1^{-\left(1-\frac{1}{n}\right)} - 1 \}$$

$$= p_1 V_1 \left\{ \left(\frac{p_2}{p_1}\right)^{1-\frac{1}{n}} - 1 \right\}. \quad . \quad . \qquad (2)$$

$$\therefore \qquad E = \left(\frac{n}{n-1}\right) p_1 V_1 \left\{ \left(\frac{p_2}{p_1}\right)^{1-\frac{1}{n}} - 1 \right\} . \ . \ . \ . \qquad (3)$$

$\therefore$ Assuming adiabatic compression for which $n = 1{\cdot}41$,

$$E = 3{\cdot}44\, p_1 V_1 \left\{ \left(\frac{p_2}{p_1}\right)^{29} - 1 \right\}. \quad . \quad . \quad . \quad . \quad . \qquad (4)$$

If V^1 is the volume of air at p_1 compressed per minute, and p_1 is in lbs. per square inch

$$\text{Work done per minute} = 3{\cdot}44 \times 144\, p_1 V^1 \left\{ \left(\frac{p_2}{p_1}\right)^{29} - 1 \right\}.$$

$$\therefore \qquad \text{Horse-power} = \frac{3{\cdot}44 \times 144\, p_1 V^1}{33{,}000} \left\{ \left(\frac{p_2}{p_1}\right)^{29} - 1 \right\}$$

$$= {\cdot}015\, p_1 V^1 \left\{ \left(\frac{p_2}{p_1}\right)^{29} - 1 \right\}. \quad . \quad . \qquad (5)$$

As a rule, p_1 is 14·7 lbs. per square inch and V^1 is then called the volume in "free air" per minute.

We, therefore, obtain the following formula :—

Horse-power required per cubic foot of free air per minute to compress to pressure p_2 lbs. per square inch in one stage

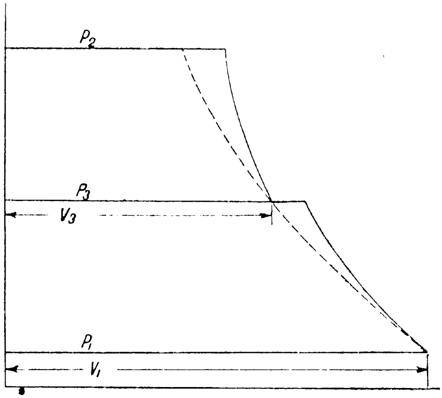
$$= {\cdot}22 \left\{ \left(\frac{p_2}{14{\cdot}7}\right)^{\cdot 29} - 1 \right\}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (6)$$

(2) *Two-stage Compression.*—Referring to the accompanying diagram, it is clear that the work done in the first stage

$$= E_1 = 3\cdot 44\ p_1\ V_1 \left\{ \left(\frac{p_3}{p_1}\right)^{29} - 1 \right\},$$

and that the work done in the second stage

$$= E_2 = 3\cdot 44\ p_3\ V_3 \left\{ \left(\frac{p_2}{p_3}\right)^{\cdot 29} - 1 \right\}$$



Now, $\frac{p_3 V_3}{\tau_3} = \frac{p_1 V_1}{\tau_1}$, and it is assumed that between the stages the air is cooled to its original temperature—*i.e.*, that $\tau_3 = \tau_1$,

$$\therefore \qquad p_3\ V_3 = p_1\ V_1.$$

$\therefore$ Total work done $= E_1 + E_2$

$$= 3\cdot 41\ p_1\ V_1 \left\{ \left(\frac{p_3}{p_1}\right)^{\cdot 29} + \left(\frac{p_2}{p_3}\right)^{\cdot 29} - 2 \right\}. \quad . \quad . \quad (7)$$

Ratio of Pressures to make Total Work a Minimum.—We have now to find the value of p_3 to make the total work a minimum.

This will be a minimum when $\left(\frac{p_3}{p_1}\right)^{\cdot 29} + \left(\frac{p_2}{p_3}\right)^{\cdot 29}$ is a minimum. Let this expression be called X.

X will be a minimum when $\frac{d\,X}{d\,p_3} = 0$.

i.e., when $\cdot 29 \left(\frac{p_3}{p_1}\right)^{\cdot 29 - 1} \times \frac{1}{p_1} + \cdot 29 \left(\frac{p_2}{p_3}\right)^{\cdot 29 - 1} \times \left(-\frac{p_2}{p_3^2}\right) = 0$

i.e., when $$\frac{p_1^{\cdot 29 - 1}}{p_1^{\cdot 29}} - \frac{p_2^{\cdot 29}}{p_3^{\cdot 29 + 1}} = 0$$

i.e., $$p_3^{\cdot 29 - 1 + \cdot 29 + 1} = p_1^{\cdot 29} \cdot p_2^{\cdot 29}$$

i.e., $$p_3^{\cdot 58} = p_1^{\cdot 29} \cdot p_2^{\cdot 29}$$

i.e., $$p_3^2 = p_1 \cdot p_2. \quad . \quad . \quad . \quad (8)$$

or $$p_3 = \sqrt{p_1 \cdot p_2}. \quad . \quad . \quad (9)$$

It will be noted that this is independent of the index n in the expansion law, and that we may express result (8) as—

$$\frac{p_2}{p_3} = \frac{p_3}{p_1}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

Moreover, $$\frac{p_3}{p_1} = \frac{p_2}{p_3} = \sqrt{\frac{p_2}{p_1}}$$

since by (8) $\frac{p_3^2}{p_1^2} = \frac{p_1 p_2}{p_1^2} = \frac{p_2}{p_1}$.

Now, putting this result into equation (7), we have--

$$\text{Total work} = 3 \cdot 14\, p_1 V_1 \left\{ 2 \cdot \left(\frac{p_2}{p_1}\right)^{\frac{\cdot 29}{2}} - 2 \right\}$$

$$= 6 \cdot 88\, p_1 V_1 \left\{ \left(\frac{p_2}{p_1}\right)^{\frac{\cdot 29}{2}} - 1 \right\} \quad . \quad . \quad . \quad . \quad (11)$$

We may, therefore, write a similar result to equation (6) as follows :—

Horse-power required per cubic feet of free air per minute to compress to pressure p_2 lbs. per square inch in two stages

$$= \cdot 44 \left\{ \left(\frac{p_2}{14 \cdot 7}\right)^{\frac{\cdot 29}{2}} - 1 \right\} \quad . \quad . \quad . \quad . \quad . \quad (12)$$

The necessary volume V_3 of the second cylinder will be given by the relation—

$$V_3 = \frac{p_1 V_1}{p_3} = V_1 \sqrt{\frac{p_1}{p_2}}.$$

(3) *x Stages of Compression.*— By exactly similar reasoning we can prove that if there are x stages of compression—

Horse-power required per cubic foot of free air per minute to compress to pressure p_2 lbs. per square inch in x stages

$$= \cdot 22x \left\{ \left(\frac{p_2}{14 \cdot 7}\right)^{\frac{\cdot 29}{x}} - 1 \right\} \quad . \quad . \quad . \quad . \quad . \quad (13)$$

In this case the ratio of volumes of successive cylinders should be $\left(\frac{p_2}{p_1}\right)^{\frac{1}{x}}$.

The following table is useful in practice for determining the horse-power required to drive compressors. A mechanical efficiency of 60 per cent. is assumed for pressures from 5 to 35 lbs. per square inch, increasing to 80 per cent. for 40 lbs. per square inch and beyond.

TABLE OF HORSE-POWERS REQUIRED TO DRIVE AIR COMPRESSORS

Single-Stage.		Two-Stage.		Three-Stage.	
Gauge Pressure in Lbs per Sq In.	Horse-power to Compress 100 cu ft Free Air per Min.	Gauge Pressure in Lbs per Sq In	Horse power to Compress 100 cu ft Free Air per Min	Gauge Pressure in Lbs per Sq. In	Horse-power to Compress 100 cu ft Free Air per Min
5	3·25	60	14·7	100	18·27
10	5·6	65	15·26	105	18·77
15	7·1	70	16·0	110	19·2
20	8·3	75	16·6	115	19·6
25	9·8	80	17·15	120	19·94
30	11·2	85	17·7	130	20·6
35	12·0	90	18·2	140	21·24
40	12·81	95	18·7	150	21·86
45	13·85	100	19·2	160	22·46
50	14·84	105	19·65	170	23·04
55	15·78	110	20·15	180	23·58
60	16·67	115	20·6	190	24·1
65	17·5	120	21·0	200	24·6
70	18·3	130	21·75	250	26·75
75	19·07	140	22·5	300	28·65
80	19·82	150	23·2	350	30·25
85	20·55	160	23·9	400	31·65
90	21·25	170	24·5	450	32·9
95	21·85	180	25·1	500	34·1
100	22·50	190	25·7	600	36·1
105	23·15	200	26·3	700	37·8
110	23·75	250	28·8	800	39·4
115	24·35	300	30·9	900	40·8
120	24·9	350	32·8	1,000	42·4
		400	34·5	1,200	44·2
		450	36·0	1,400	46·0
		500	37·4	1,600	47·7
				1,800	49·2
				2,000	50·6

Detail Description of an Air Compressor.—We are indebted to Messrs. Siebe, Gorman & Co., Ltd., of Westminster Bridge Road, London, for the accompanying description and illustration of one type of air compressor in which they specialise.

The illustration shows a sectional arrangement drawing of a two-stage belt-driven three-cylinder enclosed type air compressor.

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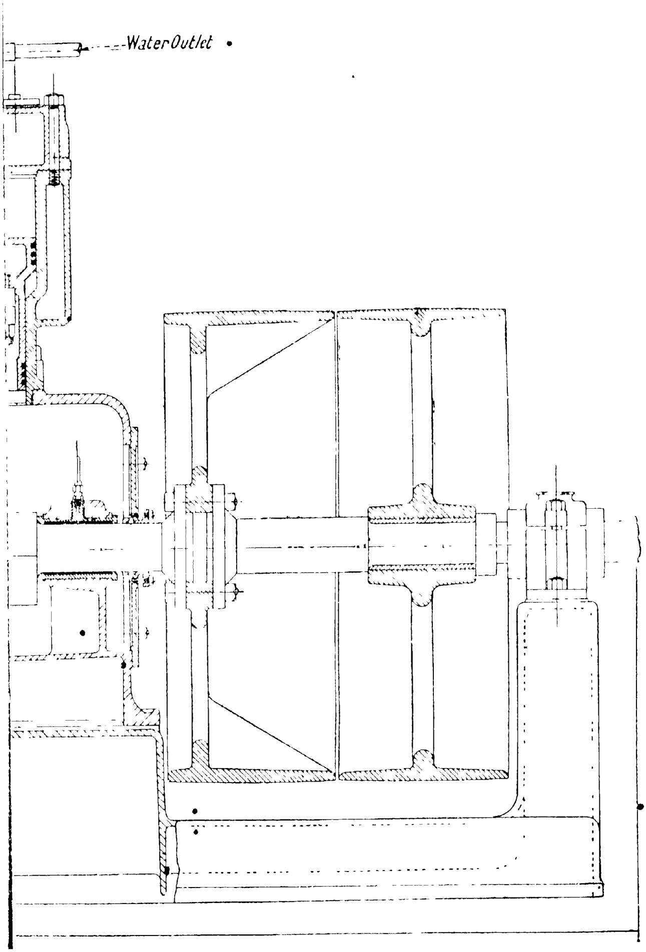


This machine is fitted with double-acting cylinders, the top portion of which constitutes the first stage and the lower annular portion the second stage. This is accomplished by the use of the differential pistons shown, the portion enclosing the annular space referred to above being so proportioned as to give the correct ratio of compression for the second-stage pressure

Before the air enters this part of the cylinder, however, it passes to the intercooler, shown in the combination base-plate, where all the remaining heat in the air of the first-stage compression is extracted. This is effected by passing the air through a series of copper pipes surrounded by water with the necessary cooling surface After leaving these pipes the air is finally compressed in the annular space, forming the second-stage compression referred to.

This compressor is fitted with forced lubrication, the oil pump being shown in the sump in the bed-plate at the left-hand end. It is driven by means of an eccentric, which in turn also drives the water-circulating pump, shown mounted on the left-hand cylinder With forced lubrication the machine is practically immune from breakdown, and can be left running with a minimum of attention

The first-stage suction and delivery valves in the cylinder head are of the disc type, made of manganese steel, having a small lift and working on renewable cast-iron knife-edge valve seats

The whole valve with its seat is very quickly removed for examination. By unscrewing the valve box cover the complete valve can be lifted out The valve boxes are in the form of pockets in the cylinder head, and they are surrounded by water, as well as the top surface of the cylinder, forming as much water-cooling surface during compression as possible

The second-stage suction and delivery valves are made of the poppet type, and are clearly shown in the cross-section of the machine. These valves are not so large as the first-stage valves, as they deal with compressed air, which has a considerably lower velocity than before it was compressed, but they have to be made stronger than the first-stage valves, as they have a good deal more racking.

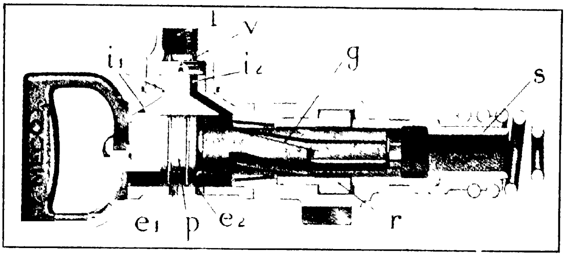
This machine can be arranged as a single-stage compressor, if desired, by connecting the air pipes in parallel instead of series, as for compounding, and, of course, doing away with the intercooler. This is accomplished by connecting all three delivery

branches together from the first-stage delivery valves, by means of a long horizontal pipe, to a similar pipe below, which connects all three poppet delivery valves together. With this arrangement the opposite suction poppet valves will now serve as a first-stage suction valve also.

The machine can be arranged steam-driven with steam cylinders mounted in tandem with the air cylinders; simple, compound, or triple expansion as desired. It also can be arranged direct-coupled to an electric motor or oil engine or driven through gearing.

Compressed Air Percussive Tools.—One of the most useful applications in practice of compressed air is for driving percussive tools for riveting, caulking, and mine drilling.

The accompanying drawing shows a cross-section through a percussive rock drill, known as the "Meco" drill, and manufactured by the Mining Engineering Company, Ltd., of Sheffield.



"Meco" Pneumatic Rock Drill.

The air enters through the pipe I and passes under the control of an automatic butterfly valve V to one or other of inlet ports i_1, i_2. In the position shown the valve is so positioned that the port i_1 is open and the port i_2 is closed; the air on the underside of the piston p is passing out through the exhaust port e_2. The piston travels towards the right, and soon covers the exhaust port e_2, and opens the exhaust port e_1. The momentum of the piston causes a compression on the underside thereof, and this compressed air passes up the port i_2, and helps to rock the valve V on its seat; this rocking is assisted by the rush of air caused by the opening of the valve e_1. The valve is carefully balanced

so that the machine will operate in any position. Immediately the valve flaps over, the compressed air passes down the inlet i_2 and causes the return of the piston, which again reverses when the exhaust port e_2 has been opened and the port e_1 closed.

The rock-drilling tool is secured in the socket s, and, in order to give a slow progressive rotation to the tool, the forward or hammer end of the piston is provided with helical grooves which co-act with a ratchet r, the co-operating pawl of which causes the socket s to give a part rotation for one stroke of the piston and to remain stationary for the next.

Lecture VII.—Questions.

1. Prove that in an air compressor which compresses 1 lb. of air at a pressure p, and volume v, to a pressure p^1, the work done is

$$p_1 v_1 \cdot \frac{n}{n-1} \left\{ \left(\frac{p^1}{p}\right)^{\frac{n-1}{n}} - 1 \right\},$$

the compression being according to the law $p\,v^n$ = constant.

2. An air compressor handles 100 cubic feet per minute of atmospheric air at 60° F., compressing it to 60 lbs. per square inch gauge pressure according to the law $P\,V^{1\cdot3}$ = constant. What is the final volume and temperature of the air, and how much power is absorbed? *Ans.* 312° F.; 12·6 H.P.

3. Show that in a compound air compressor working between extreme pressures $p_1\,p_2$, if the intermediate pressure p^1 is equal to $\sqrt{p_1\,p_2}$, the work done in each cylinder is equal.

4. An air compressor takes 100 cubic feet of free air per minute (temperature = 60° F.) at 200 lbs. gauge in two stages with intercooler cooling to 60° F. Find the size of cylinders for an expansion curve $p\,v^{1\cdot2}$ = constant. Piston speed, 300 feet per minute at 200 revolutions per minute; also find H.P. required to drive compressor, taking mechanical efficiency = 90 per cent. Clearance volume = 1/10 cylinder volume. Note, 1 lb. air at 60° F. and 14·7 lbs. per square inch has volume = 12·8 cubic feet. Also find the temperature at which the air would be delivered. *Ans.* L.P. cylinder, 9·5 inches diameter; H.P., 4·75 inches diameter; B.H.P. = 21·5; temperature, 190° F.

5. Explain why it is more advantageous to compress air in two stages with an intercooler than to use a single-stage compressor. Show from first principals that for a two-stage compressor the efficiency of the compressor will be greatest when the L.P. cylinder pressure is equal to $\sqrt{P \times P_0}$ (P_0 = original and P the final pressure).

6. Show, that if air be compressed in an air-compressor, the relation between the temperatures and pressures is given by the equation

$$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{\frac{n-1}{n}}$$

Find the temperature at the end of compression when air is compressed from 15 lbs. per square inch absolute and 70° F. (21° C.) to 105 lbs. per square inch absolute. Assume n = 1·35.

7. Show that the work done in drawing in, compressing, and discharging V_2 cubic feet of air in an air compressor from a pressure P_2 to a pressure P_1, and volume V_1 is given by the following expression:—

$$\frac{n\,(P_1 V_1 - P_2 V_2)}{n-1},$$

where the compression curve is of the form $P\,V^n$ = constant; neglect clearance. If 1,500 cubic feet of air per minute at 15 lbs. per square inch absolute pressure are to be delivered at 60 lbs. per square inch absolute, find the horse-power required. (Assume n = 1·3.)

Lecture VII.—A M.Inst.C.E Questions

1. State briefly the advantages of using multi-stage air compressors. Air is compressed adiabatically from a pressure of 15 lbs. per square inch absolute to 90 lbs. per square inch absolute. Find the final temperature of the air if the initial temperature is 60° F. (Assume $\gamma = 1.4$.)

2. Air is drawn into a compressor at atmospheric pressure and compressed to a pressure of five atmospheres. Find the horse-power required to compress and deliver 1,000 cubic feet of free air assuming (*a*) isothermal compression, (*b*) adiabatic compression. ($\gamma = 1.4$.)

3. Air, at a temperature of 80° F., is compressed from 15 to 90 lbs. per square inch absolute, and the curve of compression is $p v^{1.2}$ = constant. Find (1) the work done, and (2) the heat lost to the cylinder walls, per pound of air. The specific heats at constant pressure and constant volume are 0.2375 and 0.169 respectively.

4. An air compressor compresses the air from 15 lbs. per square inch to 90 lbs. per square inch. Find the horse-power required to compress and deliver 1,000 cubic feet of free air per minute. Neglect leakage and clearance. (Assume $\gamma = 1.3$.)

5. Show that the work done per stroke by an engine in compressing V_2 cubic feet of free air adiabatically and delivering it at a constant pressure P_1

$$= \frac{\gamma(P_1 V_1 - P_2 V_2)}{\gamma - 1},$$

where V_1 is the volume of the air at pressure P_1, and P_2 is the pressure of the air at volume V_2 (neglect clearance). Find the work required from an engine to compress adiabatically and deliver 800 cubic feet of free air per minute from 15 lbs. per square inch absolute pressure to 90 lbs. per square inch absolute pressure. (Assume $\gamma = 1.4$.)

6. Air is compressed in a two-stage compressor, the pressure at the end of the first stage being 5 atmospheres, and at the end of the second 200 atmospheres. Assuming that the comparisons are adiabatic, and that 20 cubic feet of air at atmospheric pressure and 60° F. temperature are supplied per minute, find the approximate amount of cooling water supplied at 60° F. per minute required to keep the average temperatures in the compressor constant, and calculate the horse-power required to drive the machine if its mechanical efficiency is 75 per cent.

7. An air compressor compresses 8 cubic feet of air at an absolute pressure of 15 lbs. per square inch to an absolute pressure of 105 lbs. per square inch, and delivers it to the receiver at constant pressure. Find the foot-lbs. of work done on the air per stroke (*a*) when the compression is isothermal, (*b*) when the compression follows the law $P V^{1.4}$ = constant.

LECTURE VIII.

FUELS AND COMBUSTION.

Relative Values of Fuels. In dealing with the economics of heat engines, the question of fuel is of great importance; from the point of view of the user of the engine, the most important question is cost per horse-power, and it is well for the student to remember that engines which are most economical from the standpoint of thermal efficiency are not necessarily most economical when the cost of fuel is taken into account.

The question of new sources of fuel supply has exercised the minds of engineers for some time. For purposes of light transport, motor vehicles, agricultural tractors, and like machinery, coal is being gradually replaced by oils, the light varieties of which are showing signs of shortage in supplies, and much attention has been given in recent years to the possibilities of alcohol as a fuel Assuming that the engines can be adapted to run satisfactorily on a given fuel, the ultimate question arises as to the cost of the fuel per B.Th.U., for this we must know the **calorific value** of the fuel.

The calorific value of a fuel is the number of heat units developed by the combustion of a unit weight of the fuel.

The calorific value is measured experimentally by means of calorimeters, a description of various types of which will be found in Lecture IV., Vol. I.

The following values may be taken for the calorific values of the elements and principal constituents of fuels :—

Constituent.	Atomic Weight.	Molecular Weight	Cal Value B.Th.U. per Lb
Hydrogen, H_2,	1	.	61,500*
Carbon, C,	12	.	14,500
Sulphur, S_2,	32	.	4,000
Carbon monoxide, CO,	..	28	4,320
Methane (marsh gas), CH_4,	.	16	23,500
Ethylene, C_2H_4,	..	28	21,300
Benzene, C_6H_6,	.	78	17,800

Calculation of Calorific Value of Fuel from Chemical Composition.—If we are given the chemical combustion of a fuel we can calculate its calorific value approximately by the aid of the foregoing data, but it should be remembered that accurate values can only be obtained experimentally. In these calculations it is usually assumed that of the hydrogen one-eighth of the weight of oxygen present will probably be combined with oxygen in the form of water. The following examples will make clear the method of calculation :—

Example 1.—A sample of coal gives the following composition on analysis :—C 88 per cent., H_2 3·6 per cent., O_2 4·8 per cent., ash 3·6 per cent.

$$\therefore \quad \text{Calorific value of C} = 14{,}500 \times \cdot 88 = 12{,}760$$

$$\text{,,} \quad H_2 = \left(\cdot 036 - \frac{\cdot 048}{8}\right) \times 52{,}500 = 1{,}670$$

$$\text{Total} = \text{calorific value of fuel in B.Th.U. per lb.} \quad 14{,}430$$

Example 2.—Calculate the lower calorific value per cubic foot of marsh gas (methane, CH_4), having given the density as ·0148 lb. per cubic foot.

In 1 lb. of marsh gas, the molecular weight of which is 16, there will be $\frac{12}{16}$ lbs. of carbon and $\frac{4}{16}$ lbs. of hydrogen.

*This includes the latent heat of the steam generated by the combustion and is commonly called the "Higher Value;" this latent heat usually passes out with the flue or exhaust gases, and amounts to about 9,000 B.Th.U. per lb. of hydrogen, so that for fuel calculations in practice the "Lower Value" of 52,500 B.Th.U. per lb. should be taken.

$$\therefore \text{ Calorific value of C per lb. gas} = \frac{12}{16} \times 14{,}500 = 10{,}900$$

$$,, \quad H_2 \text{ (lower)} \quad ,, \quad = \frac{4}{16} \times 52{,}500 = 13{,}100$$

$$24{,}000$$

$\therefore$ B.Th.U. per lb. = 24,000.

$\therefore$ B Th.U. per cubic foot = 24,000 × ·0448 = **1,080.**

The actual heat available for practical use is less than this, on account of the fact that in separating the gas into its constituents, carbon and hydrogen, the "heat of formation" is absorbed.

This amounts to about 110 B Th.U. per cubic foot, so that the available calorific value of marsh gas per cubic foot becomes 1,080 − 110 = 970 B.Th.U.

Solid and Liquid Fuels.

Average Composition and Calorific Value.

Description.	Specific Gravity	Percentage Composition					Calorific Value B Th U. per Lb.
		C	H_2	$O_2 + N_2$	S	Moisture	
Welsh anthracite,	..	91·5	3·5	3·4	·6	..	15,200
Nixon's navigation steam coal.	.	87·8	4·2	5·0	..	1 0	15,400
Newcastle steam coal.	..	81·3	5·3	9 9	·8	1·2	14,700
Yorkshire coking coal.	..	84·1	4·9	7·0	·6	2·2	13,400
Scotch cannel coal,	..	75·4	6·2	10·0	2·2	4·0	13,500
English coke,	..	88·4	1·4	3·3	·35	4·8	13,600
Wood (ordinary), .	..	36·4	4·6	29·6	..	28·9	5,900
Alcohol,	·81	52·2	13·0	34 8	..	..	12,600
Kerosene,	·78	85	15	..	..	..	18,900
Methylated spirits,	·92	..	..	..	..		11,000
Benzol,	..	92·3	7·7	.	.	.	18,100
Petrol,	·72	85	15	..	..	..	18,700
Crude petroleum, .	·923	86	12	..	..	..	17,900

Gaseous Fuels.

Average Composition and Calorific Values.

Kind of Gas.	Percentage Composition by Volume.							B Th U. per Cu Ft.	Cu Ft of Air required per Cu Ft. Gas
	H_2	CO	CH_4	Heavy Oils	CO_2	O_2	N_2		
Town gas, .	45	10	33	5·3	·5	·3	5·7	575	5·0
Dowson pressure.	19·8	23·8	1·3	..	6·3	.	48·8	144	1·4
Dowson suction.	13·2	25·3	·3	.	5·4	·6	55 2	122	·93
Mond pressure	16·6	27 3	3·3	..	5·2	..	47 6	165	1·4
Blast furnace gas.	2·3	24·8	·8	..	5·7	..	66·4	98	·75

Combustion Data of Gases.

Gas	Density Lbs per Cu. Ft	Specific Heat		Cubic Feet for Complete Combustion per Cubic Foot	
		Constant Pressure	Constant Volume	O_2	Air
Air, ...	·0809	·238	·169	..	
Carbon monoxide, CO, .	0784	·216	·176	·5	2·4
Carbonic acid, CO_2, .	·1225	·216	·153	.	..
Marsh gas, CH_4, . .	·0448	·593	·470	2 0	9·6
Acetylene, C_2H_2, . .	·0727	..	.	2·5	12
Olefiant gas, C_2H_4,	0784	·404		3·0	14·4
Hydrogen, H_2, . .	·0056	3·40	2·41	·5	2·4
Nitrogen, N_2, . .	·0784	·244	·173	.	..
Oxygen, O_2, . .	·0896	·218	·156	..	..

Flue or Exhaust Gas Analysis.—In order to determine the principal constituents of flue or exhaust gases, a sample of the gas is taken and the proportions of its principal constituents are determined by a gas-sampling apparatus.

The principal constituents of the gases in the flues or chimney of a boiler are as follows:—

	Symbol.
1. Oxygen, ...	O
2. Nitrogen, ...	N
3. Carbon dioxide, usually called carbonic acid gas, ...	CO_2
4. Carbonic oxide (carbon monoxide), . . .	CO

The object of the analysis is to determine the percentage of these gases present, and to deduce therefrom the amount of air actually entering the furnace, as compared with the air theoretically necessary for combustion. If all the air admitted to the furnace could be brought into such intimate contact with the fuel that every atom of the oxygen contained in it could be utilised for the purposes of combustion, the escaping gases would practically consist of only carbonic acid and nitrogen—that is, each atom of the carbon in the fuel would unite with two atoms of oxygen in the air admitted, forming CO_2, the nitrogen passing through unchanged. Such a result, however, is unattainable, and unless an excess of air be admitted, the carbon will not be completely consumed, and CO, consisting of one atom of carbon combined with one atom of oxygen, will be formed, instead of CO_2. The formation of CO results in a very serious loss of heat, and, therefore, must be prevented by admitting some excess of air.

Orsat Apparatus.—One of the most useful forms of apparatus for this purpose is the Orsat apparatus shown on the accompanying diagram, for which we are indebted to the handbook, *Steam*, issued by Messrs. Babcock & Wilcox, Ltd.

The Orsat apparatus enables the percentage of oxygen, carbon dioxide, and carbonic oxide to be ascertained directly. The remainder is usually considered to be nitrogen, as, although there are traces of other gases, they are insignificant.

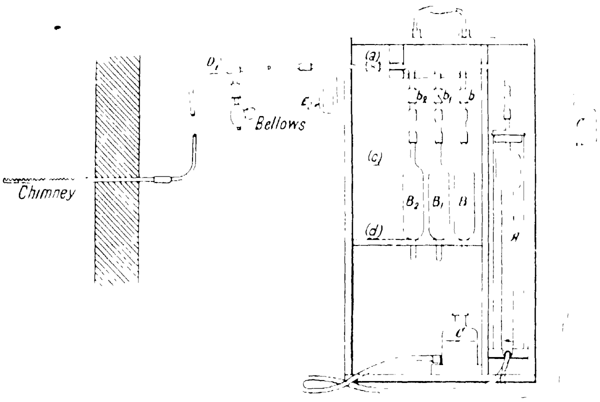
The apparatus consists essentially of a measuring tube A, into which a sample of the gas is drawn, and of three other vessels B, B_1, and B_2, which contain substances capable of

absorbing respectively, carbon dioxide, oxygen, and carbonic oxide.

The method of using the apparatus is as follows.

Through a suitable hole in the chimney, uptake, or flue, insert a piece of iron tube, long enough to reach well past the centre, the tube having saw slits in its circumferential plane for a length of 12 inches or more. If desired, a tube perforated with small holes may be used.

See that the aperture in the chimney, round the tube, is tightly plugged, so as to prevent air (which would probably vitiate the results obtained) being drawn in.



ORSAT FLUE-GAS TESTING APPARATUS.

Place the Orsat apparatus in a convenient position near the chimney, the bottom of the apparatus being, say, about 3 feet above the level of the feet of the observer, connect the outer end of the iron tube to the Orsat apparatus by an india-rubber pipe D, having a U-tube filled with glass wool inserted at the position marked E, so as to intercept flue dust.

The bottle C is to be filled about two-thirds full of water, and connected to the bottom of the measuring tube A by an india-rubber tube. When this bottle is placed on the top of the case containing the apparatus, or at some other convenient similar height, the water will naturally flow into the vessel A

If, then, the bottle C be placed below the apparatus and the cock a opened, it is evident that as the water flows out of A the gas will be drawn in from the flue and take its place. Draw in the gas well below the zero mark, and cut off the connection with the flue by closing the cock a. Then lift the bottle C, so that the water level in it exactly coincides with the zero mark in the measuring tube, and open the three-way cock a to the atmosphere to allow of the surplus gas escaping. We thus obtain the tube A full of gas at atmospheric pressure. Again close the cock a. Then, by opening one of the cocks b, b_1, or b_2, the gas contained in the measuring tube A can be forced into either of the vessels B, B_1, or B_2, by raising bottle C so that water flows into A, due care being taken that the water never rises above the mark at the top of the measuring tube. The vessels B, B_1, and B_2 contain the following reagents.

Vessel	Reagent	To Absorb
B.	One part commercial caustic potash and two parts of water (solution of sp. gr. 1·2), . .	CO_2
B_1.	Five grammes pyrogallic acid dissolved in 15 c.c. water. 120 grammes caustic potash dissolved in 80 c.c. water. The two solutions to be mixed, ...	O
B_2.	Saturated solution cuprous chloride in hydrochloric acid, ...	CO

These absorbing vessels should be filled rather more than half-way up with the reagents.

It is essential that the gas to be tested be passed through the different reagents in the order given above, otherwise incorrect results will be obtained.

The vessels B, B_1, and B_2 contain small glass tubes. These are used with the object of giving a greater wetted surface to absorb the gas introduced.

The tubes with copper wire round them are for the vessel B_2 containing cuprous chloride.

Note.—Care should be taken to keep the pyrogallic solution from air, as it absorbs oxygen rapidly. It is best to mix the potash solution with it in the tube.

The measuring tube A is, for convenience of calculation, marked off into 100 parts, so that percentages may be easily read off.

At the moment of measuring the volume of gas in the graduated tube, the water bottle must be held at such a height that the level of the water in it is exactly the same as in the graduated tube, otherwise the gas will be compressed or expanded by the difference between the two columns of water.

Before commencing the test get rid, as far as possible, of the air in the connecting tubes by using the small hand bellows shown; then draw several samples of the gas into the measuring tube, and discharge each to the atmosphere through the three-way cock *a*. Having obtained an undiluted sample, shut the cock *a*, open the cock *b*, and force the gas into the vessel B. Draw the gas back into the vessel A, and repeat the operation three or four times, so as to ensure the thorough absorption of CO_2. The last two readings on tube A should give the same result, showing the absorption is complete.

Follow the same procedure with the remaining two vessels B_1 and B_2, taking the reading of the reduced quantity of gas in the vessel A after each operation.

CO_2 is absorbed by the caustic potash solution very quickly, and it will be found that passing the gas three times through the absorbing vessel B will generally be quite sufficient. The gas, however, must be passed through the pyrogallic solution at least five or six times in order that the oxygen may be all absorbed. If this be not done the oxygen remaining will be absorbed by the cuprous chloride, and will be mistaken for CO, although there may be none of that gas present

The total of the percentages of the three gases, CO_2, CO, and O, should be about 19·5, and this rule may be used as a rough check on the analysis.

As the percentage by volume of oxygen in air is 21, the volume of air corresponding with any given volume of oxygen may be found by multiplying by $\frac{100}{21}$, or 4·762. The volume of air corresponding to a given volume of CO_2 may also be found by multiplying by the same figures.

Example.– Analysis shows . . . . . 13·5 CO_2
6 % O

Then air used for combustion	$=$	$13{\cdot}5 \times 4{\cdot}762 =$	64·3
And excess air	$=$	$6 \times 4{\cdot}762 =$	28·6
			92·9

The percentage of excess air above that which is necessary for combustion is, therefore :—

$$\frac{100 \times 28 \cdot 6}{64 \cdot 3} = 44 \cdot 4 \text{ per cent.}$$

Care should be taken with regard to the following points :—

1. The absorbent should not be forced below the point D, or some of the gas may escape and be lost, and, of course, an incorrect result obtained.

2. The absorbent must be at exactly the same level in the tube say at (*c*)- when measuring the volume after the gas has been absorbed as before.

3. Time must be allowed for the water to drain down the sides of the tube A before taking a reading. The time must be the same on each occasion, otherwise more water will drain down at one time than another, and an incorrect reading result.

Calculation of Composition of Products of Combustion from Perfect Combustion of Fuel of Given Composition.- From a consideration of the chemical formulæ representing the combustion of the various elements of a fuel, we can consider the question both from the weight and the volume point of view

Take, for instance, the case of carbon ; we have-

$$C + O_2 = CO_2.$$

This means that from the weight aspect 12 lbs. of carbon combine with 32 lbs. of oxygen to give 44 lbs. of carbon dioxide.

This formula shows that one molecule of oxygen produces one molecule of CO_2, and by Avogadro's hypothesis molecules of gases have the same volume, so that 1 cubic foot of oxygen must produce 1 cubic foot of carbon dioxide.

Take, next, the case of carbon monoxide, CO. The chemical formula gives—

$$2CO + O_2 = 2CO_2.$$

That is, 56 lbs. of carbon monoxide combine with 32 lbs. of oxygen to give 88 lbs. of carbon dioxide. Also, 2 cubic feet of carbon monoxide combine with 1 cubic foot of oxygen.

For hydrogen we have—

$$2H_2 + O_2 = 2H_2O\,;$$

i.e., 4 lbs. of hydrogen combine with 32 lbs. of oxygen and 2 cubic feet of hydrogen combine with 1 cubic foot of oxygen.

As a final example, take methane or marsh gas, CH_4—

$$CH_4 + 2O_2 = CO_2 + 2H_2O;$$

i.e., 16 lbs. methane + 64 lbs. oxygen = 44 lbs. carbon dioxide + 36 lbs. water; 1 cubic foot methane + 2 cubic feet oxygen give 1 cubic foot carbon dioxide.

The oxygen for combustion is in practice supplied in the form of air, the composition of which may be taken as—

	By Weight	By Volume
Oxygen,	23	21
Nitrogen,	77	79

∴ to produce a gas necessitating 1 volume of oxygen, we must have $\frac{100}{21} = 4·76$ cubic feet of air, and 3·76 cubic feet of nitrogen will be produced; and for 1 lb of oxygen we require $\frac{100}{23} = 4·35$ lbs of air containing 3·35 lbs. of nitrogen.

We can best explain the application of these data to calculations in practice by the aid of a numerical example

Numerical Example. *A sample of coal has 86 per cent. by weight of carbon, 4·9 per cent. of hydrogen, and 4·0 per cent. oxygen, the remainder being ash. Calculate the proportions by volume of the resulting flue gases assuming perfect combustion and the number of cubic feet of air required per pound of fuel to allow this combustion.*

Carbon.—·86 lb per lb. of fuel.

$$\therefore \text{ Weight of oxygen required} = \frac{·86 \times 32}{12} = 2·29 \text{ lbs.}$$

$$\text{Weight of } CO_2 \text{ produced} = \frac{·86 \times 44}{12} = 3·15 \text{ lbs.}$$

Hydrogen ·049 lb. per lb.

Weight of oxygen required $= \frac{·049 \times 32}{4} = ·39$ lb., and ·04 is present in the coal.

$$\therefore \text{ Additional oxygen required} = ·35 \text{ lb.}$$

In an analysis of the flue gases, the steam would condense, and would not be measured.

Altogether, we have used 2·29 + ·35 = 2·64 lbs. of oxygen, which will occur in 2·64 × 4·35 = 11·45 lbs. of air, and leave 8·81 lbs. of nitrogen.

Therefore, our flue gas will contain, for each pound of fuel burnt, 3·15 lbs. of CO_2 and 8·81 lbs. of nitrogen.

Therefore, from table on p. 127,

$$\text{Number of cubic feet } CO_2 = \frac{3 \cdot 15}{\cdot 1225} = 25 \cdot 2$$

$$\text{,, ,, } N_2 = \frac{8 \cdot 81}{\cdot 0784} = \underline{112}$$

$$\underline{\underline{137}}$$

∴ Per cent. composition by volume is given by—

$$CO_2 = \frac{25 \cdot 2 \times 100}{137} = 18 \cdot 4 \text{ per cent.}$$

$$N_2 = \frac{112 \times 100}{137} = 81 \cdot 6 \text{ per cent.}$$

Now, let us consider what the flue gas analysis would have been if there had been no hydrogen in the coal.

We should then have 2·29 lbs. of oxygen employed, producing $\frac{2 \cdot 29 \times 77}{23} = 7 \cdot 66$ lbs. of nitrogen.

∴ The flue gas will contain 3·15 lbs. of CO_2 and 7·66 lbs. of nitrogen.

$$\therefore \text{Number of cubic feet of } CO_2 = 25 \cdot 2, \text{ as before}$$

$$\text{,, ,, } N_2 = \frac{7 \cdot 66}{\cdot 0784} = \underline{97 \cdot 7} \text{ ,,}$$

$$\underline{\underline{122 \cdot 9}}$$

$$\therefore \text{Per cent. composition by volume } CO_2 = \frac{25 \cdot 2}{122 \cdot 9} \times 100 = 20 \cdot 5.$$

$$\text{,, ,, ,, ,, } N_2 = 79 \cdot 5.$$

This is practically the same composition as the air, and the difference therefrom is due to small errors introduced by approximate calculations.

Since one volume is employed to produce one volume of carbon dioxide, it will be clear that the oxygen volume in the air will be exactly replaced by carbon dioxide when the exact theoretical amount of air for perfect combustion is employed.

Calculation of Excess Air from Volume Analysis of Flue or Exhaust Gases.—There are a number of methods of computing from the analysis of flue or exhaust gases the amount of excess air provided during the combustion.

One of the most simple is the use of the relation

$$\frac{\text{Excess air}}{\text{Air required to burn the carbon to } CO_2} = \frac{\text{volume of } O_2 \text{ in gases}}{\text{volume of } CO_2 \text{ in gases}}.$$

The weight of air required to burn the carbon to CO_2 is obtained from the relation—

$$\frac{\text{Weight of carbon burnt to } CO_2}{\text{Weight of carbon burnt to } CO} = \frac{\text{volume of } CO_2}{\text{volume of } CO}.$$

This relation holds, because there is an equal weight of carbon in equal volumes of CO and CO_2.

Because we have—

$$C + O_2 = CO_2. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$2C + O_2 = 2CO. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

In (1) and (2) the same volume of O_2 produces 1 volume of CO_2 and 2 volumes of CO, and employs 1 weight and 2 weights respectively of carbon, so that in equal volumes of the two gases there will be equal weights of carbon

A numerical example will make the procedure clear.

Fuel (Weight)		Dried Flue Gas (Volume)	
C,	72 per cent.	CO_2,	12 per cent.
H_2,	3·4 ,,	CO,	·4 ,,
Ash,	24·6 ,,	O,	8 ,,
		N,	79·6 ,,

We have—

$$\frac{\text{Weight of carbon burnt to } CO_2}{\text{Weight of carbon burnt to } CO} = \frac{\text{vol. } CO_2}{\text{vol. } CO} = \frac{120}{4} = 30.$$

$$\therefore \text{ Weight of carbon burnt to } CO_2 = \frac{\cdot 72 \times 30}{31} = \cdot 70.$$

$\therefore$ Weight of air required to burn this to CO_2

$$= \cdot 70 \times \frac{8}{3} \times \frac{100}{23} = 8{\cdot}1 \text{ lbs.}$$

$$\therefore \text{ Excess air per lb. fuel} = 8{\cdot}1 \times \frac{\text{volume O}}{\text{volume } CO_2}$$

$$= \frac{8{\cdot}1 \times 8}{12} = 5{\cdot}4 \text{ lbs.}$$

Calculation of Heat Lost by Products of Combustion, etc.—In preparing a heat balance for a boiler trial, it is common to consider heat losses under the following heads :—

(1) Heat lost by products of combustion.
(2) Heat lost by excess air.
(3) Heat lost by incomplete combustion.

(1) *Heat Lost by Products of Combustion.*—To obtain this, we first require to know the *weight* of products per pound of fuel, and the method of determining this will be followed from the following treatment of the numerical example previously considered :—

Weight of air theoretically required per pound fuel

$$= 11{\cdot}6 \text{ C} + 34{\cdot}8 \text{ H}.$$

$$\left(\text{The } 11{\cdot}6 = \frac{32}{12} \times \frac{100}{23}\text{; the } 34{\cdot}8 = 8 \times \frac{100}{23}.\right)$$

$$= 11{\cdot}6 \times {\cdot}72 + 34{\cdot}8 \times 3{\cdot}4 = 9{\cdot}6 \text{ lbs.}$$

$\therefore$ Add weight of combustible in fuel $= {\cdot}75$ lb.

Total $= 10{\cdot}3$ lbs. = theoretical weight of gases.

$$\frac{\text{Weight of C burnt to } CO_2}{\text{Weight of C burnt to CO}} = \frac{\text{volume } CO_2}{\text{volume CO}} = \frac{120}{4} = 30$$

$$\therefore \quad \text{Weight of C burnt to } CO_2 = \frac{{\cdot}72 \times 30}{31} = {\cdot}70.$$

$$\therefore \quad \text{Weight } CO_2 \text{ per lb. fuel} = \frac{{\cdot}70 \times 44}{12} = 2{\cdot}57 \text{ lbs.}$$

$$\text{,, CO ,,} = \frac{{\cdot}02 \times 28}{12} = {\cdot}05 \text{ lb.}$$

$$\text{,, } H_2O \text{ ,,} = {\cdot}034 \times 9 = {\cdot}31 \text{ lb.}$$

$$2{\cdot}93 \text{ lbs.}$$

$\therefore$ Weight of N per lb. fuel $= 10 \cdot 3 - 2 \cdot 93 = 7 \cdot 4$ lbs.

$\therefore$ If t° is the temperature of the flue gases, we have from the table of specific heats on p. 127—

Heat carried away by	CO_2	$=$	$2 \cdot 57 \times \cdot 216 \times t.$
,, ,,	CO	$=$	$\cdot 05 \times \cdot 216 \times t.$
,, ,,	H_2O	$=$	$\cdot 31 \times \cdot 48 \times t.$
,, ,,	N_2	$=$	$7 \cdot 4 \times \cdot 244 \times t.$

(2) *Heat Lost by Excess Air.*— We first estimate the excess air in lbs. per lb. of fuel, in the case considered, we have already found this to be 5·4 lbs.

$\therefore$ Heat lost by excess air $= 5 \cdot 4 \times \cdot 238 \times t.$

(3) *Heat Lost by Incomplete Combustion.*— It will be seen from the figures given on p. 127 that if 1 lb. of carbon be burnt to carbon monoxide 10,200 B.Th.U. will be lost, due to incomplete combustion.

$\therefore$ Heat lost from this cause in any given case per lb. of fuel $=$ carbon burnt to CO $\times$ 10,200.

Chimney Draught.— The intensity of draught of a chimney depends upon the difference in weight of the inside and outside columns of air.

If h is the height of the chimney in feet, and A is the cross-sectional area in square feet, and τ_i and τ_e are the absolute temperatures inside and outside the chimney, then if the pressures inside and outside were the same, the weights of the two columns of air would be—

$$492 \times \frac{A\,h \times \cdot 0809}{\tau_i} \text{ and } 492 \times \frac{A\,h \times \cdot 0809}{\tau_e} \text{ lbs. respectively,}$$

because freezing point on the absolute Fahrenheit scale is 492°.

$$\therefore \text{ Draught in lbs. per ft.}^2 = 492 \times h \times \cdot 0809 \left(\frac{1}{\tau_i} - \frac{1}{\tau_e}\right).$$

Chimney draught is usually measured in inches of water, and 12 inches of water represents 62·5 lbs. per square foot.

$$\therefore \text{ Draught in inches of water} = \frac{492 \times \cdot 0809 \times 12}{62 \cdot 5} \times h\left(\frac{1}{\tau_i} - \frac{1}{\tau_e}\right)$$

$$d = 7 \cdot 6\, h\left(\frac{1}{\tau_i} - \frac{1}{\tau_e}\right).$$

A slight modification of this formula, given in Messrs. Babcock & Wilcox's handbook, *Steam*, is-

$$d = h\left(\frac{7\cdot6}{\tau_i} - \frac{7\cdot9}{\tau_e}\right).$$

The following diagram, reproduced by permission from this handbook, gives the theoretical draught power of chimneys of different heights for varying outside air temperatures at atmospheric pressure: this may be calculated by the formula given above. For example, to ascertain the draught power of

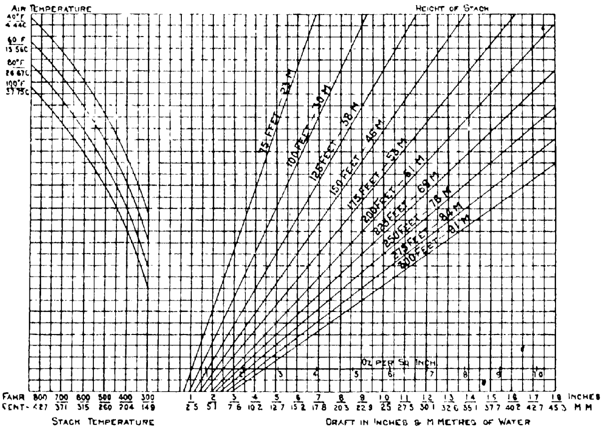


DIAGRAM OF THEORETICAL CHIMNEY DRAUGHT POWER

a chimney 200 feet high with an outside air temperature of 80° F., and with the temperature of the chimney gases at 400° F., find at the bottom of the left-hand side of the diagram the figures 400° F., follow the vertical line upwards until it intersects the 80° F. air curve, then trace this horizontal line across the diagram to the right until it bisects the 200 feet chimney line, when the draught measurements will be found indicated at the foot of the vertical line below—viz., ·98 inch. This figure gives the theoretical draught power, the actual draught power being less owing to friction and other losses inside the chimney.

Lecture VIII Questions.

1 If 26 lbs. of air are used to burn a pound of coal containing 13,500 B.Th.U., and the temperature of the flue gases is 550° F., what per cent. of heat is lost up the chimney if the temperature of the boiler-room is 70° F.? *Ans.* 23·7 per cent.

2. A coal fuel has the following percentage composition:—C 75, H_2 5, O_2 3, N_2 2, remainder ash, etc. Find the weight of air theoretically required for complete combustion, and the theoretical resulting temperature if the air actually supplied is twice that theoretically necessary? *Ans.* 10·6 lb. air; 2,600° F.

3. A flue gas analysis shows the following analysis in percentages by volume:—CO_2 12, CO 1, O_2 7, N_2 80. Find the air used in burning coal if the percentage composition of the fuel by weight is C—80, H_2—4, O_2—2. *Ans.* 15 lb.

4. Coal, known as Wayne's Merthyr coal, has the following composition (per cent.):—C 87·49, H 3·66, O 2·69, N 1·17, S ·79, ash 3·00, moisture 1·20. Find the calorific value per pound. Find also the calorific value per pound of olefiant gas (C_2H_4). *Ans.* 14,790 B.Th.U., 21,200 B.Th.U. per lb.

5. In a boiler the following analyses of coal and flue gases were obtained.

Coal (by weight)			Gases (by volume)		
C,	80·5	per cent	CO_2,	9·79	per cent
H,	4·7	,,	CO,	1·47	,,
O,	6·4	,,	O,	8·47	,,
N,	1·5	,,	N,	80·27	,,
S,	·9	,,			
Water,	2	,,			
Ash,	4	,,			
Total,	100	,,	Total,	100	,,

Find the excess air per pound and the weight carbon per pound burnt to CO and CO_2 respectively. *Ans.* 7 lbs. excess air per lb., ·70 lb. burnt to CO_2; ·105 lb. to CO.

6. In the previous question, if the inlet temperature of the air is 80° F. and the flue temperature is 580° F., find per pound of fuel the heat carried away by the products of combustion, by excess air, by moisture, and the heat lost by incomplete combustion. Compare these with the heat available in the coal. *Ans.* By products, 1,257 B.Th.U.; by excess air, 830, by moisture, 570; by incomplete combustion, 1,060; available heat = 14,130 B.Th.U. per lb.

7. The following figures are taken from the report of a trial of a four-cycle oil engine. Analysis of dry exhaust gases, by volume, per cent.:—

CO_2 12·1, CO 2·0, O_2 2·0, N_2 83·9. Composition of the oil by weight, per pound (approximate):—C 0·86 lb., H 0·14 lb. The temperature of the exhaust gases leaving the engine was 800° F. (426·5° C.), and the temperature of the air 62° F. (16·5° C.). Calculate the heat carried away by the exhaust gases per pound of oil used. The steam formed by the combustion of the hydrogen passes away as superheated steam. Specific heats of the gases per pound:—CO_2 0·216, CO 0·248, O_2 0·218, N_2 0·244.

8. Find the maximum efficiency of the generator of a suction gas producer, the composition of the gas produced, and the calorific value per cubic foot, assuming that the fuel is carbon, and that air only is passed through the fuel; given that 1 lb. of hydrogen occupies 178·8 cubic feet; that the calorific value of carbon monoxide is 342·4 B.Th.U (190·2 C.H.U.) per cubic foot; and that the calorific value of 1 lb. of carbon is 14,544 B.Th.U. (8,080 C.H.U.). What is the effect of admitting steam in addition to the air (*a*) on the working, (*b*) on the efficiency of the producer.

9. In a boiler trial 3,600 lbs. of coal were consumed in 24 hours. The weight of water evaporated was 28,800 lbs., mean steam pressure by gauge 95 lbs. The coal contained 3 per cent. of moisture and 3·9 per cent. of ash by analysis. Determine the efficiency of the boiler, and the equivalent evaporation (1) per pound of dry coal, (2) per pound of combustible. Feed water temperature 95° F. or 35° C. Calorific value of 1 lb. of coal 13,000 B.Th.U., or 7,222 C.H.U.; total heat of 1 lb. of steam at 110 lbs. per square inch absolute 1,184 B.Th.U., or 658 C.H.U.

LECTURE VIII.—A.M.INST.C.E. QUESTIONS.

1. The volumetric analysis of the flue gas of a boiler gave the following results :—CO_2, 10 per cent. ; oxygen, 9·7 per cent. ; nitrogen, 80·3 per cent. Find the weight of air supplied per pound of coal if the coal contained 82 per cent. of carbon, 5·3 per cent. of hydrogen, 4 per cent. of ash, and 8·7 per cent. of incombustible gaseous products.

2. Given the analysis of the fuel, of the flue gases, the temperature of the flue gases leaving the boiler, and the temperature of the air in the boiler-room, explain the method of calculating the heat carried away by the flue gases per pound of fuel.

3. The thermal efficiency of a gas engine working with suction gas is 30 per cent. The efficiency of the producer is 80 per cent., and anthracite of 14,000 B.Th.U. per pound is used, costing 28s. per ton. The overall thermal efficiency of a steam plant is 15 per cent., using coal of 12,500 B.Th.U. per pound, costing 12s. per ton. The brake efficiencies are 86 and 92 per cent. for the gas and the steam engine respectively. What is the cost of fuel in each case working at 100 B.H.P. for 3,000 hours, omitting stand-by losses ?

4. The consumption of steam coal costing 19s. per ton in the case of a condensing overtype superheated steam engine, is 1·4 lbs. per B.H.P.-hour when developing 200 B.H.P., and 1·9 lbs. per B.H.P.-hour when developing 50 B.H.P. A suction gas plant has a consumption of anthracite (costing 28s. per ton) of 0·9 and 1·7 lbs. per B.H.P.-hour at the same loads respectively. Draw the Willans line in both cases, and find at what load the cost of fuel for the steam plant is equal to that of the suction gas plant.

5. The thermal efficiency of a gas producer is 70 per cent., that of the engine it is supplying with gas 33 per cent., and the brake efficiency of the engine is 86 per cent. The calorific value of the fuel is 14,000 B.Th.U. Calculate the fuel consumption per B.H.P.-hour.

6. In a boiler trial the following analysis of the flue gases was obtained :—By volume—carbon dioxide, 13 per cent. ; oxygen, 6 per cent. Find the pounds of air per pound of coal if the carbon in the coal was 85 per cent. (Air consists of 23 per cent. of oxygen by weight.)

7. Explain how you would determine the carbon dioxide, carbon monoxide, and oxygen present in the flue gas of a steam boiler.

8. The analysis of the flue gases by volume for a steam boiler is—CO_2, 10·5 per cent. ; O_2, 4·9 per cent. ; CO, 0·5 per cent. ; and N, 84·1 per cent. Find the excess of air present. State what data would be required to determine the heat lost in the flue gases, and the method you would adopt for determining the loss.

9. A gas used in an internal-combustion engine had the following composition by volume :—Hydrogen, 45 per cent. ; marsh gas (CH_4), 36 per

cent.; carbon monoxide, 15 per cent.; nitrogen, 4 per cent. Find the volume of air required for the combustion of 1 cubic foot of the gas. (Oxygen in air is 21 per cent. of volume.)

10. The gases from a boiler flue pass through an economiser, entering at 650° F. and leaving at 350° F. If 4,000 lbs. of feed-water per hour are raised by this means from 50° F. to 250° F., find the weight of chimney gases passing through the economiser per hour. Specific heat of the gas = 0·24. Assuming 10 lbs. of water evaporated per 1 lb. of coal burned, find approximately the weight of air used per 1 lb. of coal burned.

LECTURE IX.

INTERNAL COMBUSTION ENGINES GENERAL THEORY.

In the steam engine, the heat generated by the combustion of the fuel is utilised to generate steam, which is the working fluid of the engine. This results in a loss of heat involved in the generation of the steam in addition to that involved in the working of the engine proper.

In an internal combustion engine the actual process of combustion takes place in the cylinder itself. As a result, it is possible to obtain much higher temperatures in the cylinder, and, as would be expected from the second law of thermodynamics, this results in greater thermal efficiency of the engine, even though it is impossible in practice fully to utilise the high temperatures involved.

The Otto or Four-Stroke Cycle.—We have already considered in Lecture IV, from the thermodynamic standpoint, the standard cycles employed in internal-combustion engines, we now will consider the cycles from the mechanical point of view.

The most common cycle employed in practice is the Otto or four-stroke cycle. It is named after Otto, who in 1876 designed the first internal-combustion engine to be adopted extensively in practice. The credit, however, for the cycle should be given to Beau de Rochas, who outlined the process in a pamphlet published in 1862; he was the first to suggest compression of the charge without the use of an auxiliary pump.

As the name implies, four strokes of the piston are required to complete the cycle: these are :—

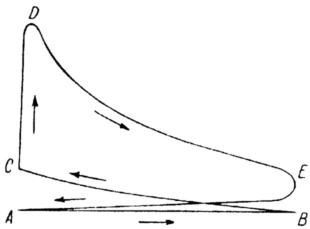
(1) *Suction stroke*, during which the piston moves forward and the charge of air and gas or atomised oil (the explosive mixture) is drawn into the cylinder.

(2) *Compression stroke*, during which the piston returns and compresses the charge into the clearance space.

(3) *Working Stroke.* The compressed charge is fired when the piston reaches its dead point In most engines this causes an explosion accompanied by very considerable increase in pressure, and the gas expands as the stroke proceeds As we shall see later, explosion occurs in all internal-combustion engines except the Diesel engine, in which combustion proceeds over an appreciable time period

(4) *Exhaust stroke*, during which the piston returns and drives the burnt gases out of the cylinder

The resulting diagram then takes the form shown



In the case of single-acting engines, and most internal-combustion engines are single-acting, it will be seen that only one stroke of the piston in four is a productive stroke, so that for at least three-quarters of the running time of the engine the parts of the engine are not doing work. Moreover, the explosion sets up high stresses, which the cylinder and associated parts of the engine must be strong enough to resist; such parts are, therefore, working at full load for only a fraction of their running time

These shortcomings of the engines do not affect their thermal efficiency, but they do affect the practical values of the engine, and many internal-combustion engines, though possessing higher thermal efficiency, are actually more expensive to run than steam engines. Internal-combustion engines are, however, more convenient in use, especially for small power.

It is due largely to the development of producer gas plants and of engines capable of working with crude oil that internal-combustion engines of large powers have come into use in recent years.

The various valves in a four-stroke cycle are usually controlled from a cam-shaft driven at one-half of the speed of the main shaft, so that each valve is operated once in two revolutions of the main shaft of the engine. Many detailed examples of four-stroke cycle engines are given later in this book.

The Clerk or Two-Stroke Cycle.—With a view to reducing the number of idle strokes in the internal-combustion engine, Sir Dugald Clerk introduced in 1881 an engine in which the explosion occurs once in every forward stroke of the piston instead of once in every alternate forward stroke, as in the Otto cycle.

A diagrammatic section,* through an air-cooled two-stroke cycle petrol engine is given herewith.

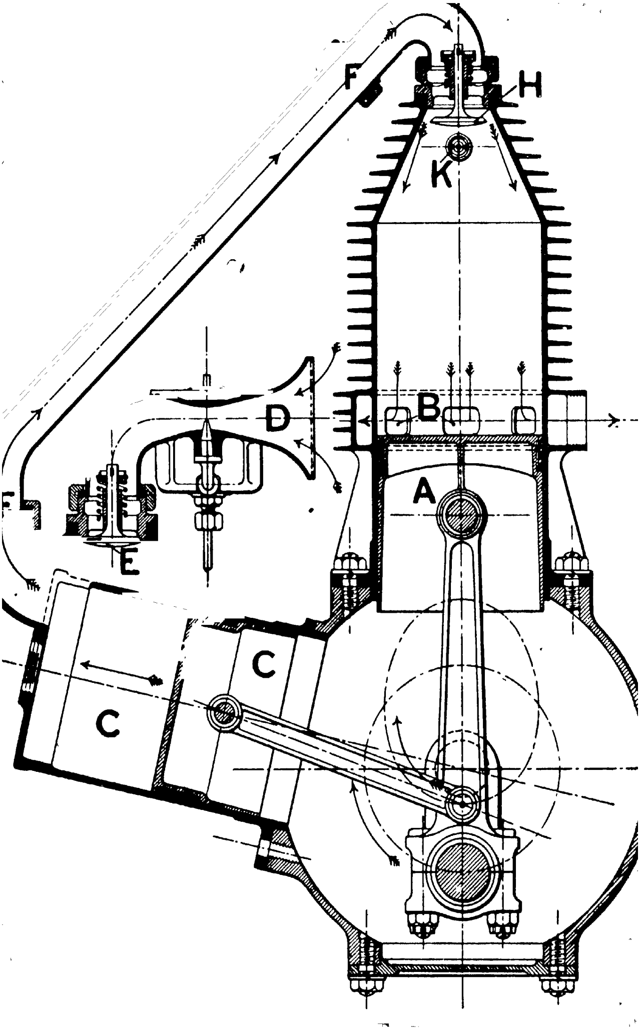
The power piston A, when near the bottom of its stroke, overruns a belt of ports B, through which the burnt gases at once escape into the atmosphere, as indicated by the arrows.

C C is a charging pump driven by the engine, by which carburetted air is drawn through the carburettor D and automatic suction valve E, and delivered through the pipe F F and automatic inlet valve H to the top of the combustion chamber.

As arranged, it will be seen that when the power piston A is near the bottom of its stroke, the pump plunger is moving rapidly towards the left, compressing its charge of carburetted air; so soon, therefore, as the pressure within the power cylinder is relieved by the exit of the burnt gases through the ports B, the superior pressure above H causes this valve to open, and fresh mixture—at a pressure of 3 to 5 lbs. per square inch, immediately flows into the cylinder, displacing the burnt gas and assisting its escape. To render this action most effectual, and at the same time to guard against loss of fresh mixture through the ports B, the combustion head is made of the expanding conical form shown in the diagram. Exhausting and charging thus occur simultaneously in the Clerk two-stroke cycle engine.

The piston on its up-stroke first covers the ports B and then compresses the entrapped fresh mixture, the valve H automatically closing as soon as compression begins, at or near the top of the stroke the compressed charge is fired by the ignition plug K, and the working stroke follows; hence every down-stroke is a working stroke, and this engine accordingly gives one working impulse per single-acting cylinder per revolution.

* This and the next illustration, together with the description, are reproduced by permission of the publishers from *Aero Engines*, by G. A. Burls, M.Inst.C.E., published by Messrs. Charles Griffin & Co., Ltd.



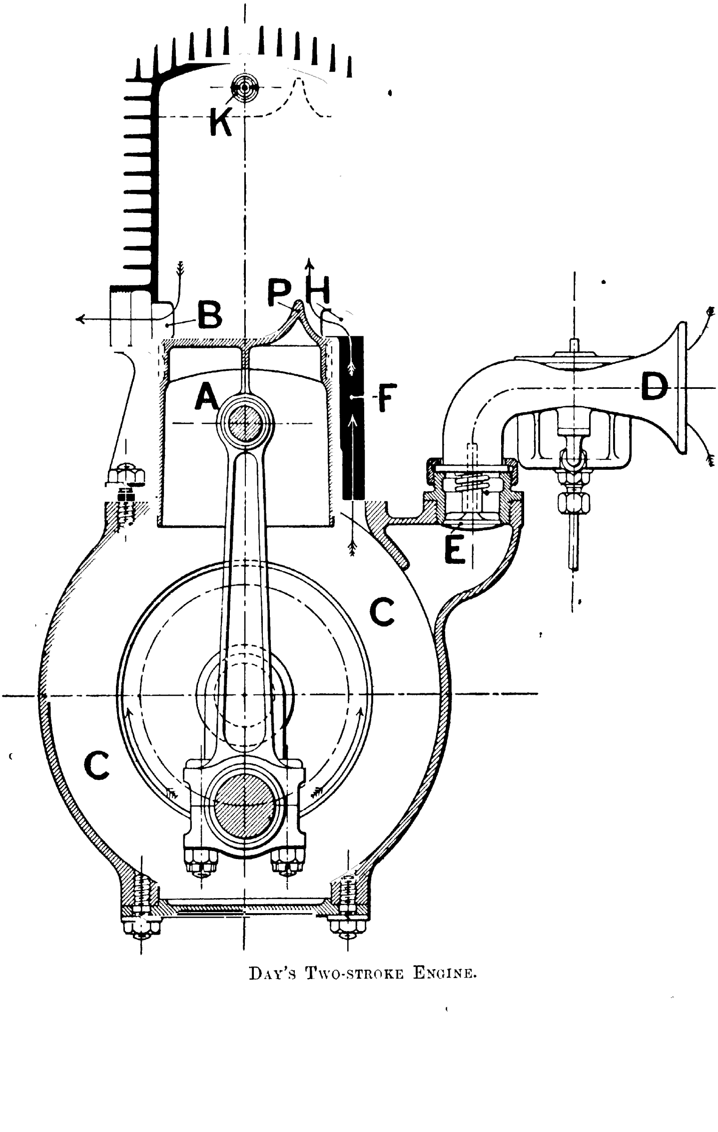
F
H
K
B
D
A
E
C
C

If the clearing-out or "scavenging" of the exhaust gases were as complete as in a four-stroke cycle cylinder, and if as great a quantity of fresh mixture could be introduced in each charge, and if the mechanical efficiency of the two-stroke engine could be made as high as that of the other, then the two-stroke engine would develop the same brake mean effective pressure as its rival, and hence at the same revolution speed would give twice as much B.H.P.

In practice, however, this is not realised; the burnt gases must escape during the short interval of time that the exhaust ports B remain open at the end of each working stroke, while the fresh charge has to enter the combustion chamber in practically the same interval; thus the scavenging is imperfect, and hence the amount of the entering fresh charge is less than in the four-stroke engine, wherein there is rather more than one complete stroke given up to each of the operations of exhausting and charging; moreover, the presence of the rather bulky charging pump reduces the mechanical efficiency of the engine.

Engines operating on the Clerk two-stroke cycle have proved very successful in the case of large stationary motors running at speeds of only 75 to 150 revolutions per minute, and many of these, of very large power, are at work, particularly in the well-known Koërting and Oechelhauser types; the defects of the cycle are, however, aggravated in the small high-speed petrol engine, and accordingly the two-stroke car or aero engine has yet to establish itself in favour. About 1912 a small quick-speed two-stroke Clerk-cycle petrol engine was produced substantially as shown on p. 146 (except that it was water-cooled), known as the "Dolphin" engine; this showed a fuel consumption at full load and 1,000 revolutions per minute of only 0·68 lb. of petrol per B.H.P. hour, with a power output estimated as 1 56 of that of an equal-sized four-stroke engine. Though a well-designed engine, the gain of power by the adoption of the two-stroke mode of working was thus only 56 per cent., while the addition of weight due to the charging pump probably resulted in the weight-power relation being increased rather than diminished.

The Day Two-Stroke Engine.—A very ingenious form of the two-stroke cycle engine was invented by Day in 1891, and is shown diagrammatically on p. 148. He dispensed altogether with a separate charging pump by causing the crank-chamber C C to perform this function, the lower part of the power piston



Day's Two-stroke Engine.

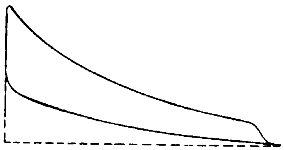
acting as the pump plunger, he contrived, moreover, to dispense with the inlet valve by arranging for the power piston itself to act as both inlet and exhaust valve. Near the bottom of the down-stroke the piston A first overruns the exhaust port B, and the burnt gases at once rush out: almost immediately afterwards the inlet port H is opened and the fresh charge under slight compression in the crank-chamber due to the descent of the piston flows into the cylinder through the passage F, and is deflected upwards by a lip P on the piston, so as to prevent, as far as possible, any short-circuiting through the exhaust B. As in the previous case, the ascent of the piston first shuts off the ports H and B, and next compresses the fresh charge into the combustion head, when it is fired and the working stroke follows, at the same time there is a partial vacuum caused in the crank-chamber and the suction valve E accordingly opens, admitting a fresh charge of mixture from the carburettor D. This is known as the "two-ported" engine, by connecting the carburettor with a port which is opened by the *lower* edge of the piston when near the *top* of its stroke the valve E can be omitted, this is the "three-ported" Day engine, and in this form the motor is valveless, though in practice it is usual to fit a simple non-return valve in the suction pipe, even in the three-ported type, to reduce the liability to "upset" the mixture.

Though the simplest form of internal-combustion engine that has yet appeared, the Day engine usually suffers in a somewhat marked degree from most of the defects of the two-stroke cycle already referred to; the imperfect scavenging causes the fresh charge taken by the cylinder to become so diluted that great care must be exercised in adjusting the carburettor to give just the requisite strength of mixture, otherwise the engine reverts to a sort of four-stroke cycle, every alternate down-stroke becoming a "scavenging" stroke, thus the Day petrol engine is very "sensitive" on its mixture. Further, there is at low speeds a considerable loss of fresh mixture by short-circuiting across through the exhaust, at high speeds this loss is much reduced, but the volume of fresh charge taken in is then much smaller. Some tests made by Watson and Fenning in 1910 on a small three-ported $3\frac{1}{4}'' \times 3\frac{1}{4}''$ Day petrol engine showed that at 600 r.p.m. 36 per cent. of the fresh charge was lost through the exhaust port, while at 1,200 r.p.m. the loss was 20 per cent. The volumetric efficiency was about 40 per cent. at both speeds, the greater loss of fresh charge through the exhaust

port at the lower speed counterbalancing the larger volume of charge then entering the cylinder.

The B H.P. was 4·2 at 1,200 r.p.m., and the mechanical efficiency at this speed was 80 per cent., the value of $n\,p$ being 51½ lbs. per square inch. Watson and Fleming concluded that this engine gave at 900 r.p.m. about 1·47, and at 1,500 r.p.m. about 1·29 of the power of a four-stroke engine of equal bore, stroke, and speed.

Notwithstanding its drawbacks, the combined advantages of simplicity, low production cost, and an impulse every revolution have resulted in very large numbers of these engines in the single-cylindered form being employed for the propulsion of small motor boats, especially in America; in recent years also a large number of builders of motor bicycles recommenced using engines of this type in the production of light and low-priced machines.



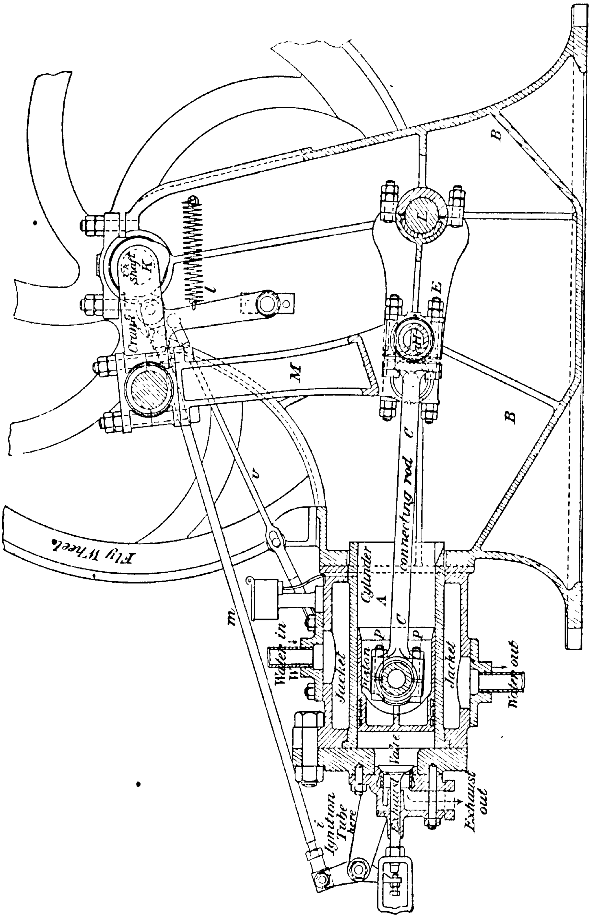
Indicator Diagram for Two-Stroke Engine.—The accompanying diagram illustrates an indicator diagram taken upon a Koërting gas engine, which is a double-acting gas engine working upon the two-stroke cycle; some further particulars of this engine are given on p. 172.

The Atkinson Engine.—Although mechanical complexity ultimately resulted in the engine becoming obsolete, the engine invented by Mr. Atkinson in 1895 was of so great interest that an account of it appears to be useful in any educational work upon the subject.

In the Otto cycle the ratio of expansion of the gases after ignition is no greater than the ratio of compression; as a result, the temperature and pressure are still appreciably high when release occurs. It is, therefore, desirable to obtain a larger ratio of expansion than compression, and Mr. Atkinson set out to design an engine in which, by the aid of a linkage mechanism, the working stroke was approximately twice the suction stroke.

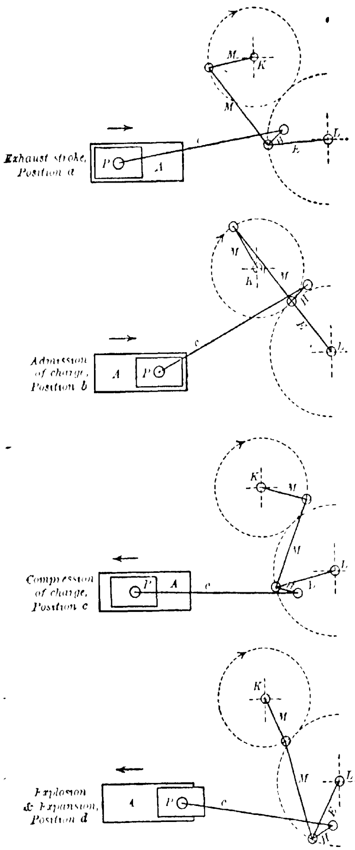
In the first form of Atkinson's engine two pistons acting differ-

entially in the same cylinder were employed; in a later form, which we illustrate herewith, the unequal strokes were all obtained with one piston, working upon the motor crank through a series



of rods, links, and levers. The admission and exhaust were operated by valves in the ordinary way.

The cylinder A is placed upon a strong base plate B, in the interior of which is mounted the linkage mechanism, four positions of which are shown in diagrammatic form on the accompanying figure.



The connecting-rod c is pivoted to a vibrating link H, which is connected by a lever E to a fixed point L, and by a lever M to the crank M^1 carried by the engine shaft K.

It will be seen from the figure that in one complete revolution of the crank M^1 the piston is given two reciprocations, the stroke on the second reciprocation being in excess of that on the first; with the proportion of links shown the ratio of stroke is 2·5 : 4·3.

A 5·5 I.H.P. Atkinson gas engine tested in London in 1888 gave a B.H.P. of 4·9 and a thermal efficiency of 19·5 per cent. reckoned on the B.H.P. This was the highest recorded efficiency at that time, but the increase in thermal efficiency was not sufficient to make up for the practical objections of complexity

of the linkage mechanism, so that manufacture of the engine was discontinued.

Scavenging in Internal-combustion Engines.—The term "scavenging" is employed in internal-combustion engines to describe the process of driving the exhaust gases out of the clearance spaces. We have already pointed out in our description of the two-stroke engine that one weakness of that engine is its poor scavenging. If the clearance spaces remain full of the products of combustion at the end of the exhaust stroke, the incoming charge will become diluted and so power will be lost.

In the early days an attempt was made to overcome this objection by the addition of two additional idle strokes to the Otto-cycle, giving a six-stroke cycle.

The cycle was, therefore, as follows:—

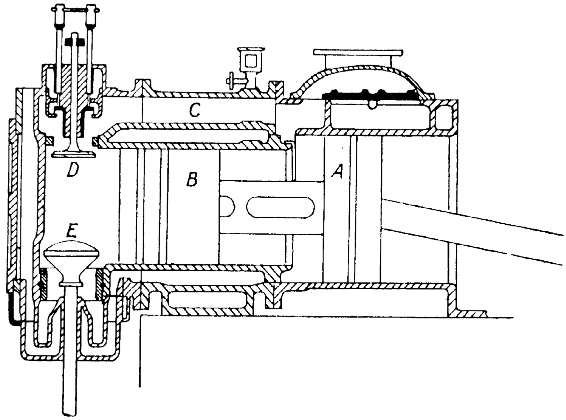
1st stroke,	. . .	Suction of charge.
2nd ,,	. . .	Compression.
3rd ,,	. . .	Working.
4th ,,	. . .	Exhaust.
5th ,,	. . .	Suction of air.
6th ,,	. . .	Scavenging by expulsion of air.

It was, however, found that there was not sufficient advantage in this procedure to compensate for the drawback of having two additional idle strokes, and the use of such strokes was soon discontinued.

A simple scavenging device introduced by Mr. Atkinson on the Crossley engines was to make use of the momentum of the steam of escaping exhaust gases to draw in fresh air, by opening the air admission valve before the exhaust valve is closed; in order to produce the necessary momentum of the exhaust gases an exhaust pipe about 65 feet long was employed. The Crossley engine with this scavenging device was found to have a high efficiency, and considerable attention was given at the time to the subject. Dugald Clerk, in his paper before the Institution of Civil Engineers in 1896, on "Recent Developments in Gas Engines," deals in detail with the trial results of this engine. He arrived at the conclusion that the procedure effectually cools the cylinder and prevents premature ignition; that the compression pressure of the fresh pure charge is increased, and with it the power of the engine. He attributed the principal

advantage to the increased compression, and this is attained in modern engines by decreasing the clearance spaces, so that special scavenging devices are now seldom employed.

The scavenging action in the Premier gas engine is obtained by the use of an enlarged end A of the piston, which is made to act as an air pump. During the exhaust stroke air is compressed into an air space C surrounding the main portion of the cylinder; this compression is caused by the difference in area of the enlarged end A of the piston and the piston proper B. When, therefore, the admission valve D is opened, the air compressed in the space C rushes in and sweeps out the burnt gases through the



Scavenging in the Premier Gas Engine.

exhaust valve E before the working charge is drawn in; for this purpose the admission valve D is opened a little while before the exhaust valve E closes.

The Explosion in Internal-combustion Engines; Theoretical Pressures and Temperatures Obtainable.—If a charge of combustible mixture of gas and air be drawn into a cylinder and the flywheel be held so that the cylinder volume must remain constant, then if the charge be ignited a rapid rise in pressure will occur. By means of the methods explained in the previous lecture, we can calculate the number of thermal units liberated by the ignition from a knowledge of the contents of the charge,

and, knowing the specific heat of the gas, we can calculate the temperature which we would expect as a result of the explosion; then assuming the law $\frac{PV}{T}$ to hold, we could calculate the resulting pressure which should be obtained if our theory is correct.

If, for instance, the heat of combustion per pound of gas is H and C_v is the specific heat at constant volume, we shall have-

$$\text{Rise in temperature} = \frac{H}{C_v}.$$

A large number of experiments of this kind have been made, and it has always been found that the pressure obtained is less than that calculated, the actual pressure being usually about one-half of the calculated pressure.

The following figures were obtained by Dugald Clerk with various mixtures of coal, gas, and air:—

Ratio Air/Gas by Volume	Pressure, lbs./in.²		Temperature °C. calculated from	
	Observed	Calculated	Observed Pressure.	Heat of Combustion
14	40	89·5	806	1,786
13	51·5	96	1,033	1,912
12	60	103	1,202	2,058
11	61	112	1,220	2,228
9	78	134	1,557	2,670
7	87	168	1,733	3,334
6	90	192	1,792	3,808
5	91	..	1,812	.
4	80	..	1,595	..

A number of explanations have been put forward to explain these observed facts which have been confirmed by other experimenters, such as Bunsen, Hirn, Mallard, and Le Chatelier.

The four chief theories that have been put forward are usually classified as—

(1) Dissociation Theory.
(2) Cooling or Wall-action Theory.
(3) After-burning Theory.
(4) Variable Specific Heat Theory.

(1) **The Dissociation Theory.** This theory is often attributed to Bunsen, and is based upon the known fact that some gases "dissociate" at high temperatures into their constituents, and that the process of dissociation absorbs heat, thus reducing the temperature and pressure of the explosion.

Experiments by Grove and Deville show definitely that steam and CO_2 commence to dissociate at a temperature of about 1,000° C. and 1,300° C. respectively.

On this theory heat will be absorbed at the high temperatures at which dissociation takes place, and will be given out again when the elements of the gas re-combine; this would cause the actual expansion curve to come above the theoretical expansion curve for adiabatic expansion, and this, in fact, is found to be the case.

On this theory, on the other hand, we should expect the effect to be more marked with rich mixtures which give a high explosion temperature than with weak mixtures which give a lower expansion temperature, whereas a study of the results obtained shows that the loss of pressure is practically the same with weak as with strong mixtures.

It is clear, therefore, that the dissociation theory does not give a complete explanation of the observed facts, and the general opinion of the scientific authorities appears to be that dissociation is only one of the contributing causes of the apparent loss of pressure on explosion.

(2) **The Wall-action or Cooling Theory.**—This theory assumes that the cooling effect of the cylinder walls is so great that the pressure actually obtained must fall much below the ideal calculated. If this theory were true, then the discrepancy should be much greater with larger vessels than with small ones, and would vary greatly with the shape of the vessel, but this is not found to be the case.

The general opinion is that wall-action is an important field for consideration, but that it does not account fully for the

observed difference between observed and theoretical results. Messrs. Bairstow and Alexander made experiments on a mixture of coal gas and air in a cylinder 18 inches long and 10 inches in diameter, the pressures being recorded upon a recording drum; they express the opinion that "at the high pressure reached by the best explosive mixtures, the loss due to cooling is less than the errors of observation, the calculated and actual values differing only on account of dissociation. At the lower pressures given by the weaker mixtures the whole of the loss of pressure is due to cooling."

The student will find an extensive treatment of the subject of wall action in Mr. A. W. Judge's book upon "High-speed Internal-Combustion Engines" (Pitman's).

In this connection we should mention that experiments have shown that the mean pressure obtained in gas engines with polished combustion chambers was perceptibly higher than in the case in which the chambers had a carbon-coated surface.

(3) **The After-burning Theory.** This theory was strongly advocated at one time by Sir Dugald Clerk to explain the observed discrepancies, he suggested that the combustion of the gas is not as rapid as supposed, and that all the heat was not liberated before the moment of highest pressure. In an actual gas engine this would mean that the gas would be burning right through the working stroke, and that it must sometimes happen that unburnt gas would pass away in the exhaust.

Experiments made by the late Prof. Hopkinson, however, led him to the conclusion that even in the weakest mixtures combustion is almost instantaneously complete when once initiated at any point, and Prof. Burstall has shown that a complete heat balance analysis can be obtained without the need of any such after-burning hypothesis.

In a later paper (1906) by Sir Dugald Clerk, he states that, "When the writer began the present investigation he believed that these phenomena of slower chemical action furnished a complete explanation of the discord between theoretical and observed results, and that there was no need to assume any considerable dissociation or variation of specific heats of the products of combustion. The present experiments, however, appear to him to indicate a real change of specific heat as well as continuation of combustion. The experiments do not exclude dissociation or any other molecular change which, by requiring the performance of work, would change specific heat."

(4) **The Variable Specific Heat Theory.**—This theory probably accounts for more of the observed discrepancy than the others which have been advanced.

After correction of the calculated results for variation in specific heats, the discrepancy reduces from about 50 per cent. to about 25 per cent. We have already dealt in detail in Chapter IV. with the effect upon the theoretical efficiency of making allowance for variation in specific heat of the gas.

The Strength of Mixture.—We have shown in the previous chapter how to calculate the quantity of air necessary for complete combustion of a fuel of given composition; experiments have shown that most economical results are obtained when there is an excess of air. For a gas requiring theoretically about 5 cubic feet of air per cubic foot of gas, Prof. Burstall obtained the most economical result with from 9·5 to 10·8 cubic feet of air per cubic foot of gas.

One explanation which has been put forward of this established fact is that with a strong mixture the composition of the gas varies in different parts of the cylinder, so that some of the gas has not around it sufficient oxygen for combustion; this would not be so likely to occur with weak mixtures. Experiments made by Prof. Hopkinson did not detect a greater percentage of unburnt gas in the exhaust from strong mixtures than from weak mixtures.

Another explanation is that with strong mixtures the temperature of explosion is higher and more heat is lost by wall action.

It is advantageous that weak mixtures give improved efficiencies because in large gas engines strong mixtures would cause excessive shocks of the explosion; with weak mixtures it is necessary to adopt high compression, but Prof. Burstall has found that in gas engines no increase in efficiency is obtained after a compression to about 175 lbs. per square inch.

Rate of Flame Propagation: Explosion.—The work of a number of experimenters upon the rates of flame propagation in mixtures of different gases, in diluted mixtures, and in mixtures under different degrees of compression, have established the following facts with regard to the rate of flame propagation in an explosive mixture:—

(*a*) It is diminished as the proportion of an inert gas present is increased.

(*b*) It depends upon the proportions of active constituents.

(*c*) It is diminished as the rarefaction of the mixture increases.

(*d*) It is increased as the temperature of the mixture is raised.

(*e*) It is much greater at constant volume than at constant pressure.

In all explosive gases there is a range of mixtures within which it is possible to obtain an explosion. In the case of petrol vapour and air, with a ratio by weight of air to petrol of less than about 8 or more than 22, it is impossible to obtain an explosion in the ordinary petrol engine.

Students requiring more information upon these important points in the theory of internal-combustion engines are referred to the following books:—Dugald Clerk's *Gas, Oil, and Petrol Engines*, 2 vols.; A. W. Judge's *High-Speed Internal-combustion Engines*; Wimperis' *Internal-combustion Engine.*

Lecture IX.—Questions.

1. A single-acting 10 H.P. air engine at 100 revolutions per minute works between 115 and 14·7 lbs. per square inch absolute pressures, works on a constant-pressure cycle, and the volume ratio of expansion is 5 . 1, neglecting clearance, and compression begins when the return stroke of the piston is $\frac{9}{10}$ completed. The expansion and compression curves are $PV^{1\cdot3} = c$. Assuming that the actual engine will give 90 per cent. of the work theoretically computed, find the size of cylinder (stroke = diameter) and the consumption of free air per I.H.P.-hour *Ans.* 11·2 inches; 645 cubic feet per I.H.P.-hour.

2. The following figures give the results of some tests upon oil engines :—

Name of Engine.	Revs. per Min.	I.H.P.	B.H.P.	Oil used	Cal value of Oil per lb B.Th.U.	Oil used in lbs per hour per I.H.P.	Oil used in lbs per hour per B.H.P.
Diesel, .	154	24·7	17·8	Light oil, .	18,370	·39	·52
Hornsby, .	202	31·0	26·7	Russolene, .	19,100	·63	·74
Crossley, .	204	20·1	15·5	Daylight, .	18,720	·61	·79
Tangye, .	200	21·4	18·0	Russolene, .	18,630	·68	·80
Priestman,	207	7·4	6·7	Russian, .	19,500	·86	·94
Premier, .	160	7·3	6·5	Russolene, .	18,600	·93	1·04

Add columns showing the mechanical efficiency, the thermal units per minute per I.H.P. and B.H.P., and the thermal efficiency taking the B.H.P. Note that the above are for different powers, and so do not give relative efficiencies of different makes.

3 Explain how the theoretical temperature obtainable from air explosive gas mixture can be calculated. Is this temperature ever realised in practice ?

4. Prove that the theoretical efficiency of an engine working on the Otto cycle depends only on the ratio of compression adopted. In an engine using this cycle 16·3 cubic feet of gas are used per I.H.P. hour, the calorific value being 650 B.Th.U. per cubic feet. If the ratio of specific heats at constant pressure and at constant volume is 1·36, and the ratio of compression is 4·5, find the actual and theoretical thermal efficiency.

5. It is invariably found, in the trial of a gas engine, that, notwithstanding the great rejection of heat to the cooling water during expansion, the expansion curve is frequently above the adiabatic. Give the different theories which have been advanced in order to explain this, quoting any experimental evidence in favour of any or all of them.

6. Give a concise account of one practical method of determining the temperature of explosion in a gas engine. In a trial of a gas engine the temperature at a point in the expansion stroke was measured and found to be 1,103° C. (2,017° F.), the corresponding pressure being 84·5 lbs. per square inch, and the volume 358 cubic inches. Find the temperature corresponding to the highest pressure of the explosion, 353 lbs. per square inch, if the corresponding volume is 130·4 cubic inches.

7. Explain the chief advantages obtained, and the difficulties to be overcome, in using a two-stroke cycle for both large and small internal-combustion engines. If the working substance may be considered to have the same properties as air, show that the efficiency E of such an engine working on either the constant pressure or constant volume cycle is expressed by

$$E = 1 - \left(\frac{v}{V + v}\right)^{\gamma - 1}$$

where V is the volume of the piston displacement, v is the clearance volume, and $p v^{\gamma} =$ constant, is the law of adiabatic expansion and compression. What is the chief error in the assumptions on which the formula is based?

8. Give a concise account of the principal results of researches to determine the values of the specific heats of gases at high temperatures. Explain what bearing these results have on the theory of the internal-combustion engine.

LECTURE IX.—A.M.INST.C.E QUESTIONS.

1. The pressure and volume at several points in the compression of a gaseous mixture in a gas-engine cylinder is shown in the following table :—

Pressure, . . .	20	45	80	130
Volume, . . .	4·73	2·52	1·62	1·10

Assume the curve can be represented by an equation of the form $P V_n = c$, and determine the value of n.

2. Describe shortly how you would carry out the test of a gas engine of, say, 20 B.H.P., using town's gas, with the object of ascertaining the gas consumption per B.H.P.-hour.

3. The explosive mixture in a gas-engine cylinder can produce 600 B.Th.U. per pound. At the beginning of compression the temperature is 250° F. Compression takes place adiabatically to one-seventh the original volume. Explosion then occurs at constant volume. What theoretical temperature would be reached, assuming constant specific heats: $C_v = 0{\cdot}19$ and $C_p = 0{\cdot}26$?

4. Sketch the indicator diagram of an Otto cycle gas engine, explaining what each portion of the diagram represents.

5. I the curve of compression or expansion in a gas-engine cylinder is of the form $P V^n = C$, explain clearly how you would determine the index n What use can be made of n after it has been determined ?

6. The following results were obtained from a small oil engine :—Revolutions per minute, 250 ; explosions per minute, 100 ; diameter of cylinder, 7 inches ; stroke of piston, 14 inches ; oil used per hour, $4\frac{1}{2}$ lbs ; calorific value of oil per pound, 18,000 B.Th.U. ; mean pressure on piston, 45 lbs. per square inch ; load on brake, 40 lbs. ; radius of brake arm, 2 feet. Find the indicated horse-power, and the mechanical and thermal efficiency of the engine.

7. A gas engine of 500 B.H.P. consumes 16 cubic feet of coal gas per B.H.P.-hour. The difference between the specific heats of the gas at constant pressure and constant volume is 120 B.Th.U., and of the mixture of air and gas after the explosion 56. The ratio of air to gas by volume is 10 to 1. The pressure at the end of the suction stroke is 13 lbs. absolute, and the temperature of air and gas at the end of the suction stroke is 100° F., the pressure at the end of compression 60 lbs. per square inch absolute, at the end of explosion 144 lbs. per square inch absolute, and temperature 2,500° F., and constant-pressure expansion takes place at that pressure until a temperature of 2,700° is reached, expansion then following the law $P V^{1{\cdot}32}$ constant. If the ratio of expansion is 4 to 1, find the clearance volume, temperature at end of compression, weight of the mixture heat, units added during explosion (at constant volume), and temperature of release assumed to be at the end of the stroke, and heat rejected in the exhaust.

LECTURE X.

GAS ENGINES.

CONTENTS.—Ignition; Slide Valve; Hot Tube; Electric—Governing Gas Engine; Hit-and-miss Method; Variable Quality Method—The Campbell Gas Engine—The Koerting Gas Engine.

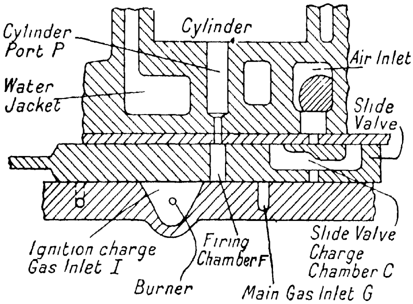
WE will now describe in some detail the construction of some gas engines and their accessories. Nearly all gas engines of small and moderate powers work upon the Otto cycle (four-stroke), but some large power engines working on the Clerk (two-stroke) cycle give very good results in practice.

The first form of gas engine to obtain extended use was the horizontal single-acting type, which is still by far the most common; Otto's engine of 1876 possessed many of the features—the open cylinder, trunk piston, and half-speed cam-shaft—of the familiar form of horizontal engine, and was first manufactured in this country, we believe, under Otto's patents by Messrs Crossley.

Ignition.—In the 1876 form of the Otto engine the ignition of the compressed gases was effected by carrying a tongue of flame, through a narrow port in a transverse slide valve, from a gas jet that was kept burning outside. To prevent the gases from blowing back the valve, when the explosion took place the slide was arranged so that the port in it, which served to carry the flame, had passed under a cover which shut it off from the atmosphere, before it reached the fixed port in the cylinder cover through which the flame passed to ignite the contents of the cylinder. This method is now quite obsolete.

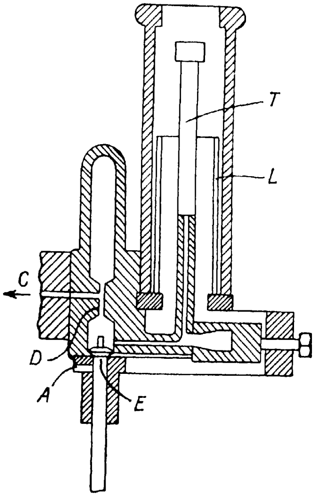
In the position shown the firing stroke is taking place; in the position of the slide valve on the extreme left the firing chamber is in register with the ignition charge gas inlet I, and the cylinder on the suction stroke sucks in a charge of gas and air through the slide valve charge chamber C, which is then in register with the cylinder port P. The slide valve then moves to the right as compression takes place, and the firing chamber F, then full of gas, passes the burner and the gas becomes ignited,

and the slide immediately moves to the position shown, whereupon the flame extends to the compressed charge.



FLAME IGNITION OF OTTO'S ENGINE.

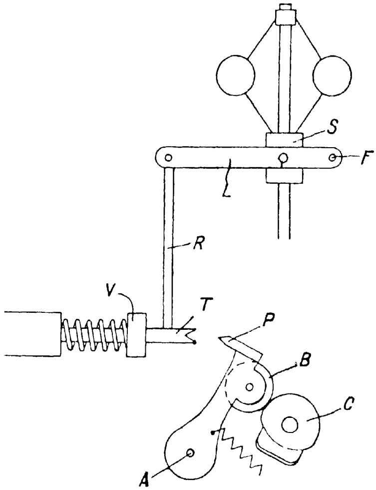
The Hot Tube Ignition.—This slide-valve method of ignition was replaced in 1888 by the hot-tube ignition device, which can still be found giving satisfactory results on some gas engines. A cast-iron tube T is closed at the top, and is surrounded by an asbestos lining L; it is kept at a red heat by a Bunsen burner. During the compression stroke a cam on the counter-shaft lifts the timing valve E into the port D. No portion of the compressed charge can, therefore, enter the tube, and any burnt gases left in it escape through a port A into the atmosphere. At the inner dead point, when the piston has completed the compression



HOT-TUBE IGNITION FOR GAS ENGINE.

stroke, the valve E is withdrawn and the compressed gas passes through the port C into contact with the red-hot tube T, and is there fired and ignites the charge.

Electric Ignition.—Electric ignition is now gradually replacing the older methods of ignition in many gas engines. As in the case with petrol engines, the electric ignition was at first obtained by means of a high-tension spark from an induction coil, but low-tension magneto ignition is now most common.

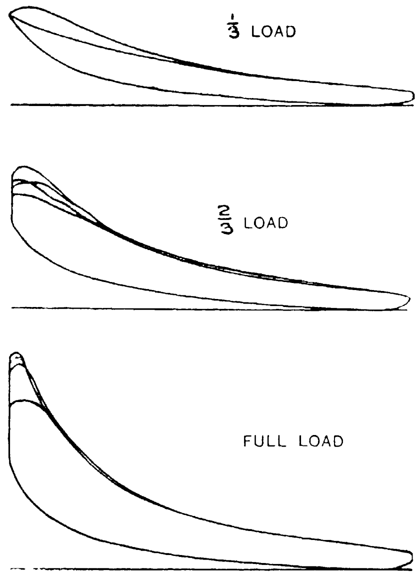


"HIT-AND-MISS" GOVERNING OF GAS ENGINES.

Governing Gas Engines.—The most common method of governing gas engines is by the "*hit-and-miss method.*" In this method of governing, the operation of the gas inlet is effected through a transmitter controlled by a governor, either of the centrifugal or the inertia type. If the speed rises, the transmitter is moved out of the operating "pecker," so that the gas-inlet valve does not become operated; the engine, therefore, misses explosions until the speed reduces to normal.

The figure illustrates in diagrammatic form this method of governing. The transmitter T is suspended by a rod R from a

lever L fulcrumed at the end F; this lever is also pivoted to the sleeve S of the governor; in the normal position of the governor the transmitter hangs centrally with respect to the valve V. The pecker P is mounted on an arm A provided with a roller B, which is held by spring action in contact with a cam C; this pecker engages the transmitter T in the normal position of the latter, and thus operates the gas valve V. When the speed rises, the sleeve S lifts the transmitter T out of the path of the pecker, so that no operation of the gas valve results.



INDICATOR CARDS FOR GOVERNING BY VARIABLE QUALITY.

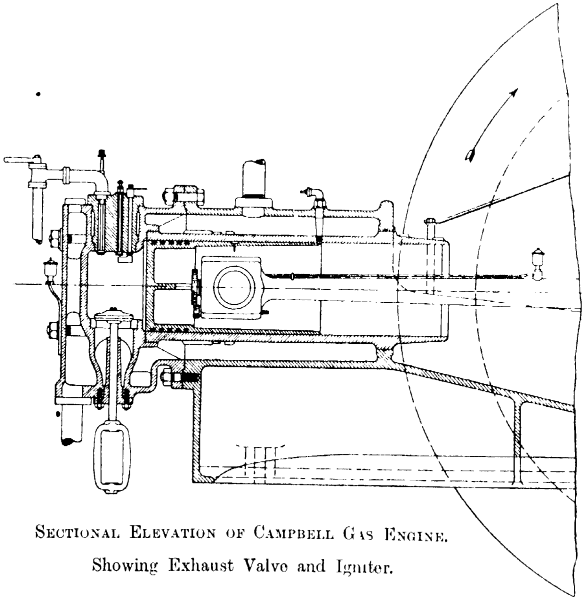
Where a more sensitive method of governing is required, so as in engines for driving electric generators, the *variable quality* method is often employed.

In this method of governing the compression remains practically constant, and the piston receives an impulse every cycle. The variable quality of the charge is usually secured by opening

the gas valve earlier, or later during the suction stroke, but always closing it at the end of the stroke.

The accompanying diagrams were taken from a Campbell gas engine, and illustrate the effect of variable quality governing upon the indicator cards.

An advantage of the constant compression which is clearly brought out by the cards is that we obtain a constant cushioning for the inertia of the reciprocating parts.



SECTIONAL ELEVATION OF CAMPBELL GAS ENGINE.

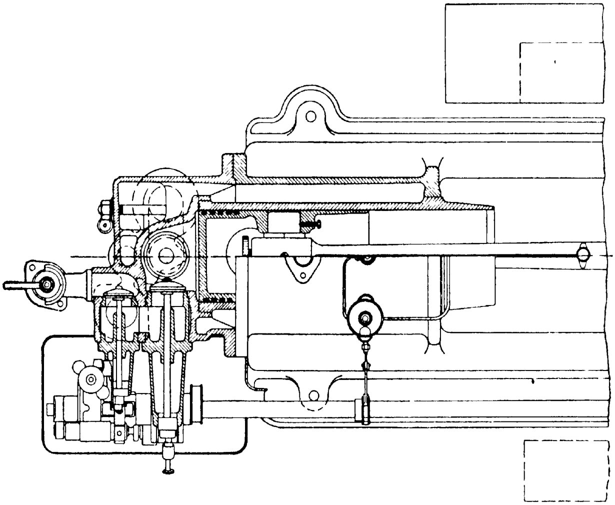
Showing Exhaust Valve and Igniter.

The Campbell Gas Engine.—The accompanying drawings of a Campbell gas engine giving 35 B.H.P. on producer gas and 41 B.H.P. on town gas illustrate the detail construction of a good design of gas engine. The following particulars, quoted from the specification issued by the Campbell Gas Engine Co., Ltd., of Halifax, may be of interest.

Bedplate.—A feature of the bedplate is that it extends beneath the cylinder as far back as the combustion chamber, thus sup-

porting the cylinder along the whole working length. Most of the engines have an oil trough around the bedplate; this catches all waste oil and prevents it from soaking into the foundation, and it also largely increases the area of the bedplate resting on the foundation.

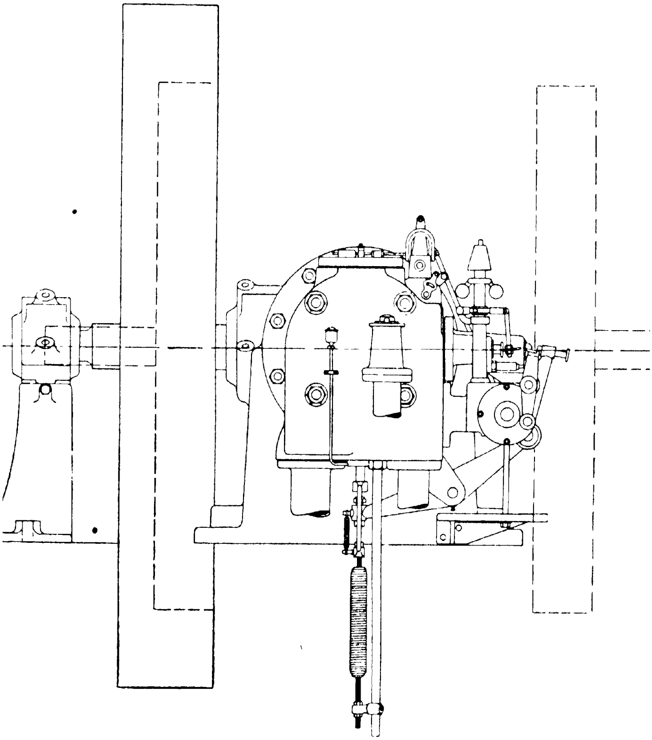
Cylinder.—In all cases a removable line of specially hard iron is provided. In the smaller engines the whole cylinder is separate from the bedplate casting; in the larger engines the combustion chamber containing all the valves, etc., is removable.



SECTIONAL PLAN OF CAMPBELL GAS ENGINE.
Showing Air and Gas Valves.

Crank-shaft, Connecting-rod, and Gearing.—These are machined from solid forgings of the best Siemens-Martin acid open-hearth steel, the material used being subjected to periodical tests. The crank-shaft runs in long adjustable bearings, and in larger sizes is fitted with balance weights. The connecting-rod has adjustable marine pattern bearing at the large end, the small end being solid with asjustable brasses. In engines of 85/100 max. B.H.P. and upwards the gudgeon pin is shrunk into the end of the con-

necting-rod and works in two adjustable bearings in the piston. All gear wheels are machine-cut and provided with suitable guards.



END ELEVATION OF CAMPBELL GAS ENGINE.
Showing " Hit-and-miss " Governor.
(The dotted lines show alternative construction with two flywheels).

Valves.—All gas and mixture valves are fitted in removable plugs or valve boxes with ground metal-to-metal joints requiring no packing. Exhaust valves have covers with ground joints and

removable guides. All cams and rollers are accurately machined and rollers run on hardened steel pins.

Ignition.—Magneto electric ignition is fitted to all the standard gas engines. The smaller engines up to 9/10 max. B.H.P. inclusive, are fitted with the high-tension system. Engines above this size are fitted with low-tension magnetos, and also with a variable timing device which can be operated while the engine is at work.

Lubrication.—In most of the engines the piston and cylinder are lubricated by a force pump, with sight feed, easily regulated while at work, and an independent sight feed lubricator is provided for the piston-pin bearing, so that it is not dependent upon receiving oil from the cylinder. The crank-pin bearing is fitted with a centrifugal oiler having sight feed which can be regulated and replenished while the engine is running. The crank-shaft and cam-shaft bearings are fitted with automatic ring lubricator; the oil is contained in an enclosed reservoir, and is used over and over again.

Horse-power.—The horse-powers specified are those which can be obtained over long periods; they are based upon an engine working at about sea level with a producer gas of calorific value 135 B.Th.U. per cubic foot or town gas of 600 B.Th U. per cubic foot.

If an engine is installed at any appreciable height above sea level, less power will be developed owing to atmospheric conditions. The loss of power is about 3 per cent. for every 1,000 feet (300 metres) elevation. This applies to any internal combustion engine.

Fuel Consumption.—The consumption of anthracite coal varies from $\frac{3}{4}$ lb. to $1\frac{1}{4}$ lbs. (·34 to ·56 kg.) per brake horse-power per hour, according to the size of the engine. The consumption of coke is about 25 per cent. greater, and of charcoal about 30 per cent. greater. An engine working with town gas will use from 14 to 22 cubic feet (·40 to ·62 cubic metre) per B.H.P. per hour. A large engine uses less fuel per horse-power than a small one.

Cooling Water.—The standard water tanks are of suitable capacity for a temperate climate like that of England. In hot countries greater cooling capacity is required, amounting to two or even three times as much as that necessary in a temperate climate.

With the larger engines, especially in a hot climate, it is often advisable to instal some form of cooler which will occupy considerably less space than tanks of the necessary capacity.

PARTICULARS OF CAMPBELL SINGLE-CYLINDER GAS ENGINE.

Producer Gas Brake Horse-Power. Maximum Load	Producer Gas Brake Horse-Power. Working Load	Town Gas. Brake Horse-Power. Maximum Load	Town Gas. Brake Horse-Power. Working Load	Revolutions per minute.	Dimensions of Standard Pulley Dia. Ins	Dimensions of Standard Pulley Width Ins	Dimensions of Standard Flywheel Dia. Ft	Dimensions of Standard Flywheel Dia. Ins	Dimensions of Standard Flywheel Width Ins.	Standard Water Tanks. No.	Standard Water Tanks. Height Ft	Standard Water Tanks. Height Ins	Standard Water Tanks. Diam Ft	Standard Water Tanks. Diam Ins	Approx Overall Dimensions of Engine only Length F	Approx Overall Dimensions of Engine only Length Ins	Approx Overall Dimensions of Engine only Width Ft	Approx Overall Dimensions of Engine only Width Ins
4	3¼	4½	3¾	320	10 ×	5	2	10 ×	4½	1 ..	5	0 ×	2	0	5	1 ×	3	4
6	5	7	6	320	12 ×	6	3	4 ×	4½	1 ..	5	6 ×	2	6	5	7 ×	3	8
9	7½	10	8½	280	18 ×	8	4	0 ×	5	1 ..	5	9 ×	2	9	6	5 ×	3	11
11½	10	13	11	280	21 ×	8	4	3 ×	5	1 ..	6	0 ×	3	0	6	10 ×	4	2
14	12	16	14	260	24 ×	10	4	6 ×	6	2 ..	5	6 ×	2	6	8	4 ×	5	1
16	14	18	15½	250	27 ×	10	5	0 ×	7	2 ..	5	9 ×	2	9	8	7 ×	5	2
18	15½	21½	18½	240	27 ×	12	5	3 ×	7	2 .	5	9 ×	2	9	9	4 ×	5	6
21½	18½	25	21	220	30 ×	12	5	6 ×	7	2	6	0 ×	3	0	9	6 ×	5	6
25	21	29	25	210	33 ×	12	6	0 ×	9	2	6	0 ×	3	0	10	0 ×	6	9
30	25	34	29	210	36 ×	12	6	6 ×	9	3 ..	5	9 ×	2	9	11	0 ×	7	1
35	30	41	36	210	42 ×	14	6	9 ×	9	3 ..	6	0 ×	3	0	12	0 ×	7	3
42	37	49	42	200	48 ×	16	7	0 ×	12	3 ..	7	0 ×	3	0	12	3 ×	8	0
50	43	58	50	200	48 ×	16	7	0 ×	12	4 ..	7	0 ×	3	0	12	3 ×	8	0
58	50	68	58	180	54 ×	18	7	6 ×	12	4 .	7	0 ×	3	3	13	9 ×	9	3
65	56	77	65	180	60 ×	20	8	0 ×	14	4 .	7	6 ×	3	6	14	0 ×	9	6
70	60	84	70	180	60 ×	20	8	0 ×	14	4 ..	7	6 ×	3	6	14	3 ×	9	6
85	70	100	85	180	66 ×	22	8	0 ×	16	5 .	7	0 ×	3	3	15	0 ×	10	9
96	80	113	96	170	66 ×	22	8	6 ×	18	5 .	7	6 ×	3	6	15	3 ×	10	10
108	90	126	108	160	72 ×	24	8	6 ×	20	6 .	7	6 ×	3	6	16	0 ×	11	9
120	100	142	120	160	78 ×	24	8	9 ×	21	5 .	8	0 ×	4	0	16	3 ×	12	0
130	110	153	130	160	78 ×	24	9	0 ×	20	5 .	8	0 ×	4	0	17	3 ×	12	6
156	130	185	156	150	84 ×	28	9	6	22	6	8	0 ×	4	0	17	6 ×	13	0

Compressed Air Starting Apparatus.

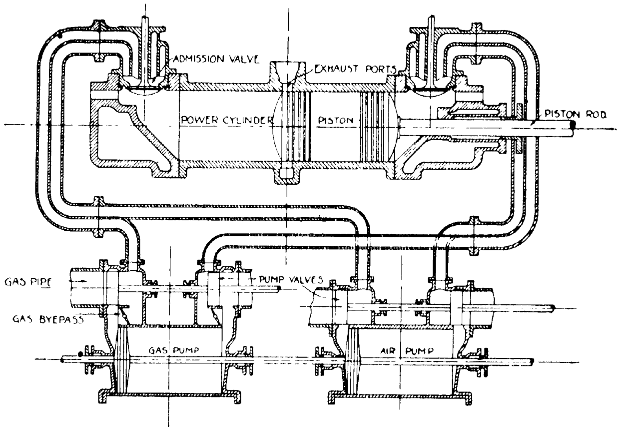
Description.	Maximum B.H.P. of Engine for which suitable	Maximum Working Pressure.	Usual Revs. per Min.	Pulleys. Dia Wdth
		Lbs. per sq. in.	About	Ins Ins
Self-contained starter, having belt-driven air compressor mounted upon the air receiver, ...	56/68 B.H.P.	150 lbs.	250	24×3½
Belt-driven air compressor, with separate air receiver, 4′ 0″ high × 1′ 6″ diameter, ...	85/100 B.H.P.	200 lbs.	250	24×3½
Belt-driven air compressor, with separate air receiver, 5′ high × 2′ diameter, .	156/185 B.H.P.	200 lbs.	250	24×3½
Air compressor direct coupled to gas engine, with separate air receiver, 5′ high × 2′ diameter. Do. do., but with paraffin (kerosene) engine. Do. do., but with benzine (petrol) engine. Do. do., but with electric motor.	156/185 B.H.P.	200 lbs.		

The Koërting Gas Engine.—The Koërting engine, which is manufactured in England by Messrs. Mather & Platt, Ltd., is an interesting example of a large power gas engine working on the two-stroke cycle. Engines of this type are extensively used as blowing engines, utilising blast-furnace gas, and they have been made in sizes capable of developing 2,000 H.P. in a single cylinder.

The cylinder is double-acting, and there are two working impulses per revolution, the charge of gas and air being delivered to the engine cylinder by separate pumps. The engine is of simple construction, and all working parts are above floor level.

The figure shows a horizontal section through the power cylinder and the two pump cylinders, one of which supplies air

and the other gas to the power cylinder. The length of the power cylinder is practically equal to twice the stroke, and has more or less conical ends, at which are situated the admission valves. Round the centre of the cylinder are situated a number of exhaust ports communicating with an exhaust belt leading to the exhaust pipe and silencer. The piston is double-ended, with convex ends, and in length equals the stroke. When the piston is nearing the end of its stroke—as in the figure (it is moving to the right)—the exhaust ports are uncovered, allowing the products of combustion to escape to the atmosphere. At



KOERTING GAS ENGINE.

the end of the stroke, when the cylinder pressure has fallen to that of the atmosphere, the admission valve is opened and a scavenging charge of air is forced through by the air pump, thus effectually clearing the cylinder. The gas pump—the discharge of which is slightly later than that of the air pump—now forces a charge of gas into the cylinder, which, mingling with the air from the air pump, forms the mixture for the power stroke. As the piston moves backwards the exhaust ports are covered, and the charge is compressed and ignited in the usual manner. The same cycle is repeated at each end of the piston

alternately, and thus the turning moment approaches that of a steam engine.

The gas and air-charging pumps are placed alongside the engine and driven from the crank-shaft by a crank about 90° in advance of the main crank. They are provided with mechanically operated piston valves, and deliver the charges at a pressure of 3 to 5 lbs. per square inch. The mixture is regulated by the gas bye-pass valves shown, which are operated by the governor, so that more or less gas is returned to the suction pipe according to the load on the engine.

The mechanical efficiency of the engine is relatively high (about 84 per cent), considering that power is required to operate the pumps.

LECTURE XI

PETROL ENGINES.

CONTENTS.—Introduction—Carburetters—"Zenith" Carburetter—Detail Description of Typical Petrol Engines—25 H.P. DORMAN ENGINE—General Data and Description; Lubrication; Cylinder Construction; Crank-shaft and Case; Flywheel and Cone Clutch; Power Curve—360 H.P. ROLLS-ROYCE AERO ENGINE; Some General Considerations on Aeroplane Engines; General Data and Description of Rolls-Royce Engine; Lubrication and Cooling; Starting Gear; Performance.

PETROL is the name given in Great Britain to a light distillate of crude petroleum, the specific gravity at 60° F. varying from ·70 to ·76, it is not a definite chemical compound, but a mixture of paraffin consisting chiefly of hexane (C_6H_{14}), heptane (C_7H_{16}), and octane (C_8H_{18}), the boiling points respectively of which are 155° F., 208° F., and 248° F.

Petrol is known in America as "gasoline," in France as "essence," and in some other Continental countries as "benzine."

Petrol engines are used principally for traction purposes, and in considering the design and efficiency of motor car engines we have to remember that the thermodynamic efficiency is sometimes of relatively small importance. The first essential is reliability in running and ease in starting; so great has been the improvement in the last twenty years that the present generation has almost to rely upon the music-hall performer for an idea of the difficulties with which the pioneer motorists had to contend.

Initial cost is a consideration which affects many people's choice of engines of proved reliability; the other standpoint upon which most motorists judge engines from the point of view of efficiency is the number of miles which the car will run to the gallon of petrol. This, of course, depends largely upon the power of the engine, the design of the car, upon the wind resistance, and several other factors.

Moreover, by the horse-power of petrol motor car engines is usually meant the R.A.C. rating based upon the formula—

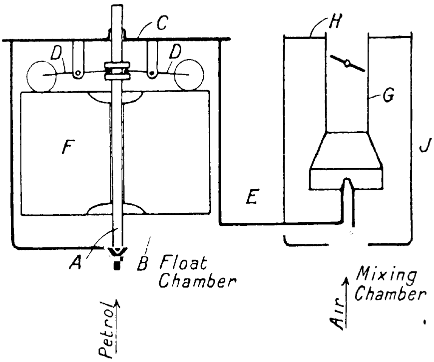
$$\text{Nominal B.H.P.} = \cdot 4\, d^2\, N,$$

where d is the diameter of the cylinder bore and N is the number of cylinders.

This horse-power is employed for purposes of taxation, and bears little relation to the actual brake horse-power developed by the engine.

Carburetters.—The appliances employed to atomise the petrol and to supply it to the engine with the requisite amount of air are called carburetters. We have seen already that the requisite amount of air in the resulting gas fuel depends upon a large number of factors, which vary under running conditions in practice.

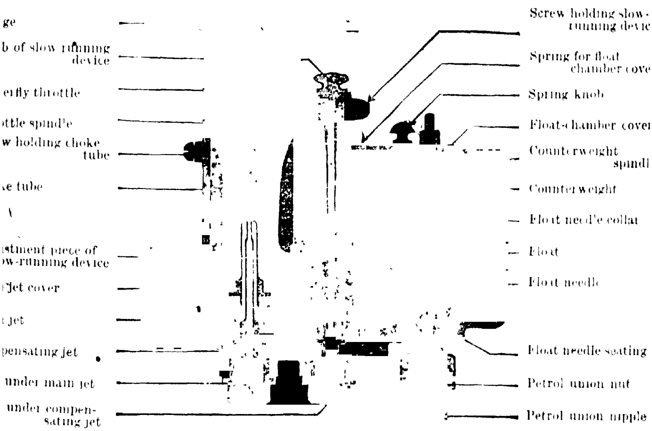
The type of carburetter in most common use may be designated as the float-feed, spray-nozzle type.



The accompanying figure shows in diagrammatic form the principal components of most carburetters of this type.

The carburetter has two main parts—the float chamber and the mixing chamber. The petrol passes in through a tube provided with a conical valve seating B, with which co-acts a needle valve A, the upper end of which usually projects above the chamber, in order that it may be manually operated. A collar C, carried by the valve spindle, engages weighted levers D, which normally hold the valve open, but which, when engaged by the upper surface of a closed hollow float F, press the collar C downwards, and this drives the valve A down upon its seat, thus interrupting the flow of petrol; by this means petrol is automatically maintained at constant level in the float chamber.

A passage E, opening into the float chamber, is connected to an upturned pipe provided with a narrow nipple J; surrounding this nipple is a funnel-shaped member G, called the choke-tube, the adjustment of the conical portion of which relative to the nipple determines the available annular area for the passage of air. The tube is so positioned that the cross-section of minimum valve is at the level of the top of the nipple, as the induced air then moves with its maximum velocity at this position and more thoroughly mixes with and vaporises the fuel. Between this tube and the induction pipe of the engine to which it is attached



ZENITH CARBURETTER—SECTION.

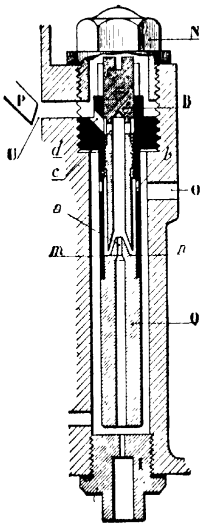
by a flange connection or union nut, is the throttle valve H, which determines the volume of mixture admitted to the cylinders, and which is operated by levers by the driver in the case of a motor car engine.

In the suction stroke of the engine a fall of pressure is caused around the jet, with the result that petrol is sucked out and a draught of air sucked in at the same time.

"Zenith" Carburetter.—The "Zenith" carburetter, which has very extended use, is of the general type previously described, but has a special compensating jet and a slow-running device

for use when the engine is running at zero load, particularly at starting.

Returning to the description of the general type of float carburetter, it will be seen that as the engine increases in speed the suction becomes stronger, and thus the amount of petrol drawn in per stroke increases, and it is found that the mixture becomes richer as the speed increases, whereas for perfect running and maximum power at all speeds the mixture should be as constant as possible.



ZENITH CARBURETTER—ENLARGED DETAIL OF SLOW-RUNNING TUBE.

To overcome this tendency for richer mixtures at high speeds, some compensating device has to be introduced, and in the Zenith carburetter this device consists of a compound jet, comprising a main jet and a compensating jet, the former being fed direct from the float chamber without any compensation whatever. Consequently, as the engine speed increases, the mixture given by the main jet grows richer and richer, and to correct this the compensating jet produces the opposite effect, because it feeds into a chamber which is open to the atmosphere, so that the discharge is unaffected by the suction of the engine.

It will be seen from the accompanying illustration that the petrol through the compensating jet passes into the central chamber, which is open to the atmosphere near the top; the petrol thence flows into the tube surrounding the main jet, and the level of this petrol will remain constant whatever the suction of the engine. As the suction of the engine increases the amount of air drawn in will increase, so that the mixture from the compensating jet becomes weaker. For starting or slow-running, the central tube, which is a small carburetter in itself, is employed.

The slow-running tube dips into the central well and is connected to a small bye-pass hole opening into the carburetter proper at the edge of the throttle valve. Consequently, by opening the throttle a shade and cranking the engine over, an

intense suction is set up at this bye-pass hole, and thus a suitable mixture is drawn in. This device is adjusted for any particular engine by altering the relative positions of male and female cones A and G; the knob B is turned to the left or right, and when the most suitable position is found it is locked in position by the nut X.

In each case the mixture is made richer by turning the adjusting piece to the right, whilst it becomes weaker by turning it to the left. Therefore, in order to get easy starting and slow running, it is only necessary to try the adjusting piece Q or B, as the case may be, in various positions, and when the most suitable is found it is fixed there.

Detail Description of Typical Petrol Engines—25 H.P. Dorman Engine.—In order to give the student a fair knowledge of the detail construction of a typical petrol engine of good design, we will now describe in detail the 25 H.P. Dorman engine, manufactured by W. H. Dorman & Co., Ltd., of Stafford There are, of course, very many good makes of petrol engines for motor-car use, and authorities differ among themselves as to the best detail arrangement of some of the parts.

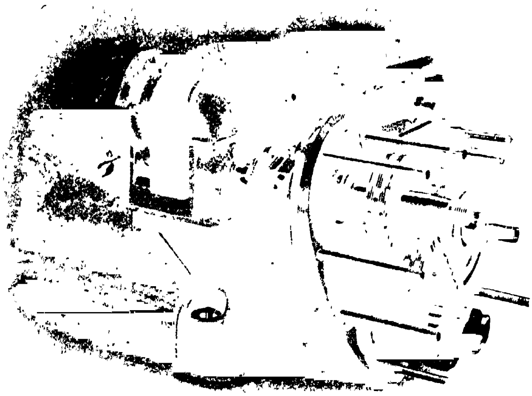
General Data and Description.

Four cylinders,	Bore 95 mm. ($3\frac{3}{4}''$).
	Stroke 140 mm. ($5\frac{1}{2}''$).
Volumetric capacity of cylinders,	3,970 c.c.
R.A.C. rating,	22·4 H.P.
B.H.P.,	20 at 800 revs. per min.
	25 at 1,000 ,,
	35 at 1,500 ,,
Petrol consumption on full load,	·7 pint petrol per B.H.P.-hour.

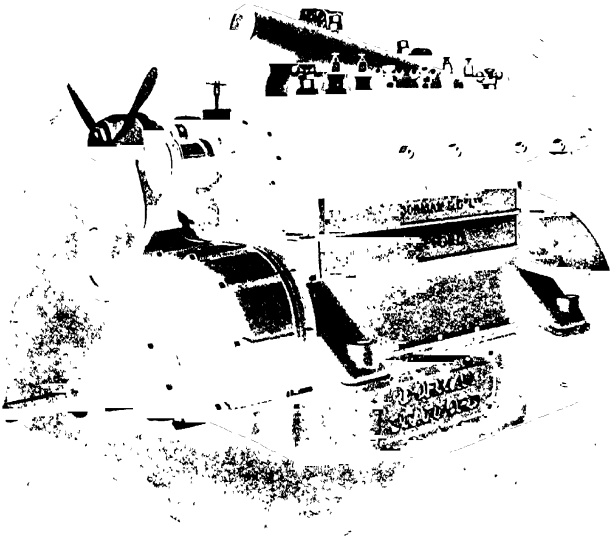
B.H.P. running on paraffin (kerosene) may be taken as 20 per cent. below the above figures.

We will first give a general description of the engine, and will then describe some of the portions in detail.

The ignition is given by a magneto, which is placed at the side of the engine, as shown in the illustration on p. 180, and is

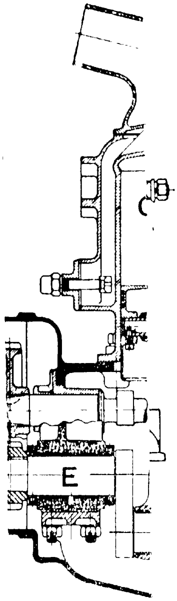


25 H.P. Engine. Chain Drive and Flexible Coupling for Magneto.



25 H.P. Dorman Engine—Perspective View.

25



Lon
25 H -P.

shaft.

Rods.
gs of Mainshaft.

driven by chain gearing from the main crank-shaft A. The pistons B of the four cylinders C are connected by connecting-rods D to the crank-shaft, the cranks on which are arranged in one plane; the crank-shaft A is provided with end bearings E and a central bearing F. The valves G are of the mushroom type, and are operated in sequence by cams on the cam-shaft H driven by chain gearing from the shaft A. The oil-pump I for the forced lubrication system is driven from the cam-shaft. The exhaust gases pass into an exhaust manifold J, which is ribbed to promote cooling and is secured by clamps which allow for expansion when heated. The starting handle-shaft K is provided with a claw-clutch L, and the flywheel is secured to a flange M on the crank-shaft.

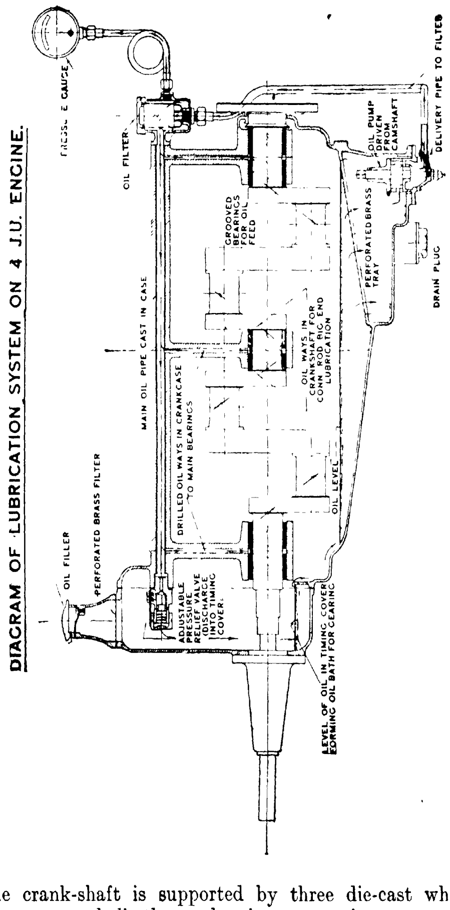
A removable cover N is provided to give access to the valve tappets.

Lubrication.—The accompanying illustration shows the progress of oil through the forced feed cycle. The oil pump is located at the lowest point of the oil sump, and is driven by spiral gearing from the cam-shaft. A large strainer on the suction side is so placed that all solid matter is excluded from the pump, and all impurities are collected in a well, which can be washed out by removing a plug; on the delivery side an oil filter is provided as an additional safeguard against dirt reaching any bearing.

Cylinder Construction.—The cylinders are cast in monobloc form, with large water spaces, especially around the valve ports and between each pair of cylinders, the whole block being capped by a one-piece water riser. This riser, which embodies a large diameter pipe leading at an angle to connect with the radiator intake pipe, is readily detachable in order to facilitate removal of any "fur" that may have been deposited in the interior of the jackets from the use of hard water.

The cylinder boxes are ground in a special machine after rough boring, and are finished to a half-thousandth of an inch. The interiors of the cylinders are tested to a pressure of 500 lbs. per square inch, and the water jackets to 50 lbs. per square inch.

Crank-shaft and Case.—An aluminium crank case, divided horizontally, is employed. The lower half forms an oil sump, while the crank-shaft, cam-shaft, and distribution gear are carried by the upper half; the latter has four integral bearer arms for supporting the engine in the frame or under frame of the car.



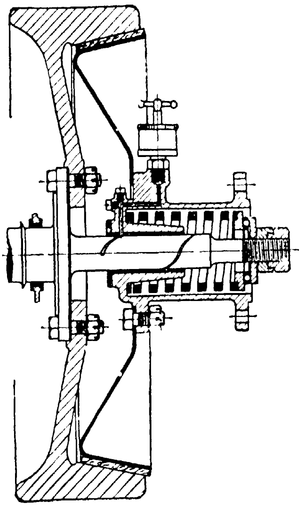
LUBRICATION OF 25 H.P. DORMAN ENGINE.

The crank-shaft is supported by three die-cast white-metal bearings, two ball thrust bearings occurring respectively in front and behind the foremost journal; it is drilled for the

lubrication of the crank-pins and journals under pressure, and has an oil thrower behind the end of the rear bearing to prevent leakage, the lubricant which passes out of the back end of the bearing being returned to the sump before it can reach the capped end of the crank-case.

The Valve Gear.—The valves themselves are of a special alloy steel, and operated in extremely long guides formed with the cylinder block and passing through the water spaces. Sparking plugs are accommodated in the inlet caps, while in each exhaust cap a compression and priming cock is provided.

The single cam-shaft has integral cams, which are ground on their operative surfaces after hardening to ensure a profile that will provide silent operation and a large area of opening. The tappets are of unusually large diameter, and each has a long bearing in its guide formed in the crank case, but excessive inertia is obviated by drilling each from the top end, leaving only sufficient metal at the bottom to form an adequate thickener for the mushroom-ended foot. The top end is threaded internally to receive a hollow and hexagon-headed stud, which forms the means of adjustment for valve clearance and makes contact with the valve stems.

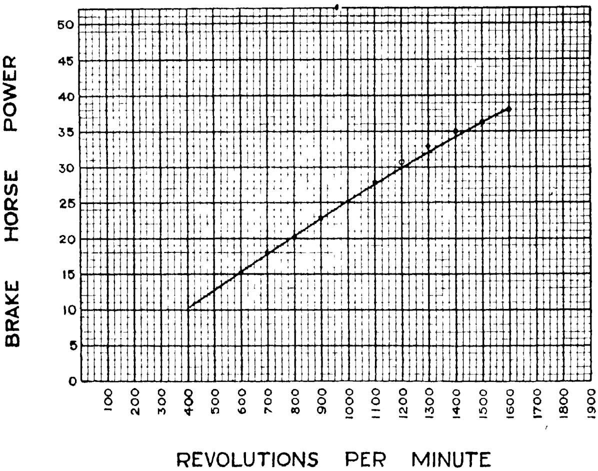


Clutch and Flywheel Detail, 25 H.P. Dorman Engine.

To retain the valve springs, each valve stem is threaded at its lower end, to receive an internally threaded cup, which is prevented from rotation by a split pin passing through inverted castellations in the cup and a hole in the stem.

Flywheel and Cone Clutch.—The flywheel is of cast iron, and is secured to the crank-shaft flange. The clutch cone is a steel pressing, lined with "Ferodo."

Power Curve.—The accompanying diagram shows the power curve for this engine; it will be seen that the B.H.P. is approximately proportional to the speed.



CURVE SHOWING VARIATION OF BRAKE HORSE-POWER WITH SPEED FOR 25 H.P. DORMAN ENGINE.

360 H.P. Rolls-Royce Aero Engine *Some General Considerations on Aeroplane Engines.*—We will now give a brief description of the 360 H.P. Rolls-Royce aero engine, which has been one of the most uniformly successful of the recent aero engines. (See frontispiece for illustration.)

In connection with aeroplane engines, we may remind the student that it was the development of a light engine that made flying by aeroplane a practical possibility.

The progress of flight has been so rapid that we can hardly realise the tremendous advances which engine design has made. A 75 H.P. horizontal oil or gas engine of usual design weighs roughly 15,000 lbs.; several 75 H.P. aero engines have been used weighing less than 200 lbs.

The following table is quoted from Mr. G. A. Burl's excellent

book on *Aero Engines*,* to which the student is referred for detailed information of the leading types of successful aero engines.

AVERAGE WEIGHTS OF INTERNAL-COMBUSTION ENGINES.

Type.	*Lbs. per B.H.P.*
Stationary 1-cylinder Diesel engines,	600
,, 2-cylinder ,,	500
,, 3-cylinder ,,	350
Marine Diesel engines,	200 to 300
Horizontal 1-cylinder oil (kerosene) engines,	300
,, ,, coal gas engines,	150 to 200
Specially light high-speed Diesels for submarines,	60
Four-cylinder petrol or kerosene engines for motor boats,	50 to 80
Four-cylinder petrol engines of motor cars,	16 to 25
"Aero" petrol engines, 4 to 20 cylinders,	2½ to 6

In addition to the necessity of low weight per horse-power in aero engines, we have to pay great attention to reliability in running. If a motor car develops engine trouble, we can pull up at the side of the road and examine the engine— if necessary on our back, while our friends are tendering amateur advice from a comfortable position but if an aeroplane engine develops engine trouble a forced landing must be made, with the attendant risk of accident.

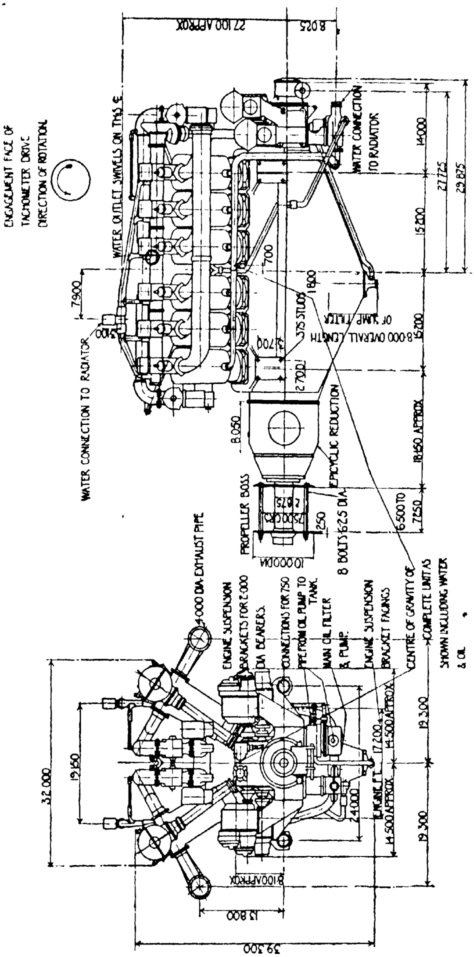
General Data and Description of Rolls-Royce Engine.—The engine has twelve cylinders, and is of the V-type—*i.e.*, the cylinders are arranged in two groups place at an angle with each other, the angle being 60°. This form of engine has a very good balance of rotating parts, and gives a uniformity of turning moment that is better than that in any other type.

The bore of the cylinders is 4·5 inches, and the stroke of the piston 6·5 inches.

GENERAL DATA.

Normal B.H.P.,	330
Normal engine speed,	1,650 revolutions per minute.
Normal propeller speed,	990 ,, ,,
Speed which engine should not exceed,	1,800 ,, ,,
Fuel consumption at normal power and speed, using Shell "B" petrol,	21 gallons per hour.
Weight of engine, including propeller hub, hand-starting gear, carburetters, magnetos, engine feet, reduction gear, but excluding exhaust process, radiator, oil, fuel, water, starter motor and battery,	900 lbs.

* Chas. Griffin & Co., Ltd.



General Arrangement and Leading Dimensions of Rolls-Royce 360 H.P. Aero Engine.

The speed reduction from the crank-shaft to the propeller is through an epicyclic gearing with a ratio of ·6: the position of this gear is given on the general arrangement diagram, and the makers claim that the epicyclic gear is preferable to any other type of reduction gear, because it causes no pressure on the crank-shaft bearings due to reaction of the drive, seeing that the direction of motion is not reversed. The axial arrangement of this gear also enables economy in the casing to be effected, as many of the heavy stresses are avoided.

The cylinders are made of forged steel, and ignition is effected by four six-cylinder magnetos, two firing on each side of the engine, and each cylinder being provided with two ignition plugs.

Lubrication and Cooling.—The lubrication system is of the type in which one oil-pump supplies pressure oil to the main bearings, and other parts, while one scavenger pump evacuates the accumulation of oil in the crank-case to the oil tank, which should be provided in the installation. Each oil pump is protected by a strainer, which can easily be detached and cleaned.

The oil consumption is approximately 1 gallon per hour. The quantity of water carried in the cylinder water jackets, water pipes, and pump is 3·1 gallons. The water system is completed to three points, at which pipes or couplings joining to the radiator may be fixed; these consist of one inlet connection to the water pump, and two outlet connections, one for each group of six cylinders.

Starting Gear.—For starting the engine an electric motor is fitted, arranged to turn the engine at about 25 revolutions per minute through a reduction gear of ratio 100 : 1, after the induction pipes have been "primed" with petrol; the explosions in the cylinders are brought about by the operation of a hand magneto. The control gear for operating the starter finishes with a wire pulley on the engine, so that connection can be made to it to a distance, if desired.

The special priming device is a light and simple apparatus embodying a hand pump, which can be fixed in any convenient position near the pilot's seat: one priming device may serve two or more engines with the use of a change-over cock.

A detachable handle is incorporated for "cranking" the engine by hand through part of the reduction gear, through which the electric motor drives.

Performance.—The accompanying chart gives the power output of the engine and fuel consumptions at various speeds.

The data upon which the chart were plotted were obtained by coupling the engine to a dynamometer and measuring the brake horse-power at certain rates of fuel consumption and various speeds. During all these tests the mixture regulator was set to give the best mixture, the necessary adjustments

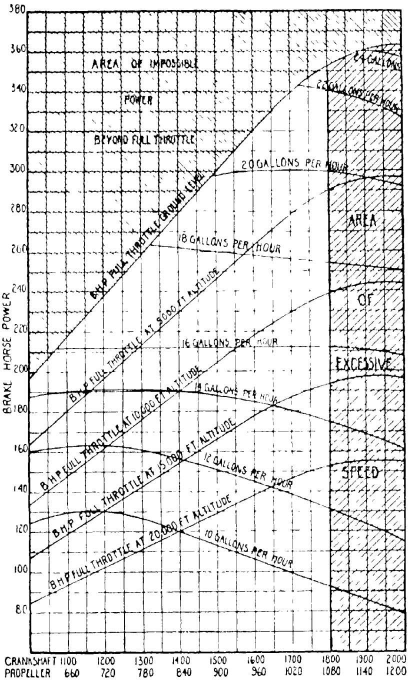


CHART SHOWING B.H.P. OF ROLLS-ROYCE AERO ENGINE AT FULL THROTTLE OPENING AT VARIOUS ALTITUDES AND B.H.P. AT VARIOUS SPEEDS AND FUEL CONSUMPTIONS AT GROUND LEVEL.

being made by altering the dynamometer setting and varying the throttle opening.

The chart, as well as that shown on p. 184, shows the great speed elasticity which is now possible with the petrol engine; in the early days of petrol engines the range of speed at which the engine could run while giving out an appreciable amount of power was very small. As a result, it was necessary for the driver continually to be operating the change-speed lever in making a journey in a motor car.

LECTURE XII.

OIL ENGINES.

CONTENTS.—The National Oil Engine; General Description; Vaporiser; Cylinder and Valves—Principal Overall Dimensions—Instructions for Working—Use of Heating-up Lamp—Starting-up by Hand; Water Injection—Stopping the Engine.

STRICTLY speaking, of course, a petrol engine is an oil engine, and so is a Diesel or semi-Diesel engine, which we shall consider in the next chapter. The kind of oil engine which we will consider in this chapter is the slow-running engine similar to the gas engine, which burns ordinary comparatively light oil, such as paraffin or kerosene.

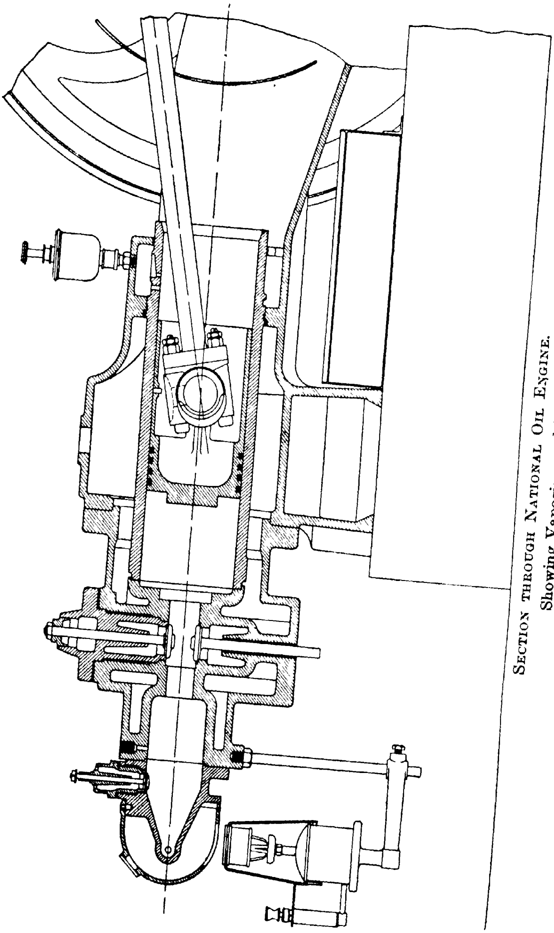
In an oil engine the carburetter of the petrol is replaced by the *vaporiser*, which serves to combine the oil and air into an explosive mixture.

The National Oil Engine.—The accompanying illustration of the oil engines manufactured by the National Gas Engine Co., Ltd., Ashton-under-Lyne, shows in detail the construction of a typical oil engine of good design.

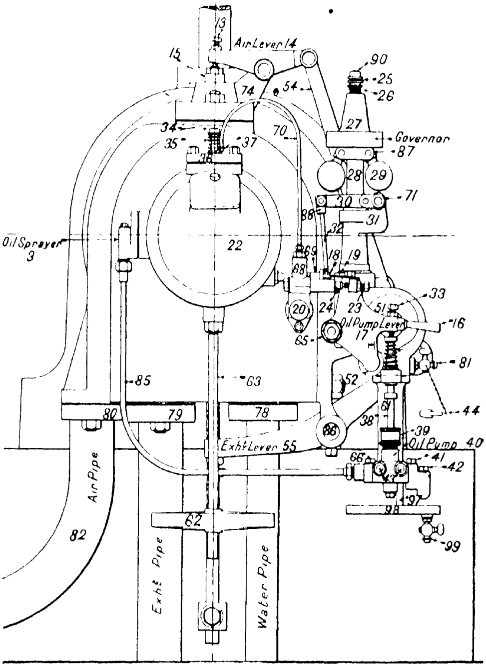
General Description.—This oil engine, as is the case with practically all similar engines, works on the four-stroke cycle. The governing is on the hit-and-miss method described in general detail on p. 165. [In the illustration on p. 192 the governor die or transmitter 18 will be noted, and also the pusher blade or pecker 19.] The bed is massive and of girder form, with a continuous arm in the line of direct stress, thus providing for strength and rigidity, and it extends well to the rear of the engine so as to reduce the overhang of the cylinder.

Water injection is employed to prevent overheating of the vaporiser at full load, and to assist in maintaining the cylinder clean.

Vaporiser.—The vaporiser, which is shown clearly on the sectional view, is of the type in which a heating lamp is required for a few minutes only during starting, the explosions afterwards generating sufficient heat to effect future vaporisation and ignition.



Section through National Oil Engine.
Showing Vaporiser and Lamp.



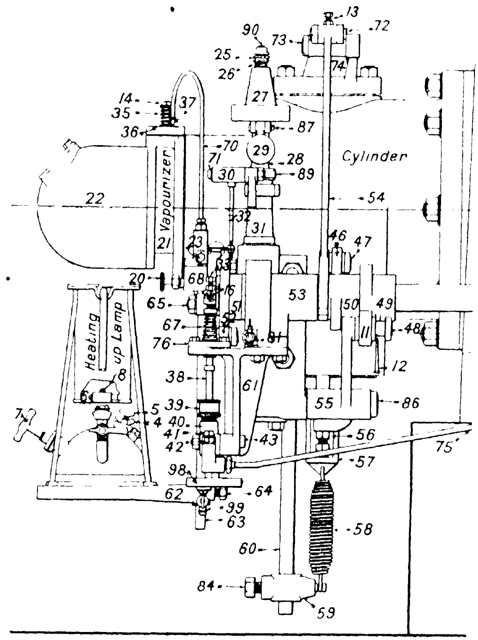
NATIONAL OIL ENGINE. END ELEVATION

DESCRIPTION OF PARTS.

3. Oil Sprayer
4. Screw Cap
5. Air Relief Cap
6. Heating-up Dish
7. Air Pump
8. Nipple for Oil Spray

Heating up Lamp

11. Exhaust Roller.
12. Exhaust Roller Relief Pin
13. Air Lever Adjusting Screw
14. Air Lever.
15. Air Valve
16. Oil Pump Lever Handle
17. Oil Pump Lever.
18. Governor Die.
19. Pusher Blade
20. Water Cock
21. Vaporiser
22. Vaporiser Hood
23. Water Pump Adjusting Screw
24. Water Pump Plunger
25. Speed Adjusting Nuts.
26. Governor Spring
27. Governor Weight
28. Governor Sleeve
29. Governor Balls
30. Governor Lever.
31. Governor Stem.
32. Governor Hanging Link
33. Oil Pump Adjusting Screw
34. Water Injection Valve.
35. Spring (Water Injection Valve)
36. Water Injection Valve Box.
37. Water Delivery Pipe Nozzle
38. Oil Pump Plunger
39. Gland for Oil Pump
40. Oil Pump
41. Oil Pump Valve Cover
42. Oil Pump Valve Cover
43. Oil Pump Supporting Bolts.
44. Oil Pump Stop
46. Air Roller
47. Air Roller Pin
48. Exhaust Roller Pin.
49. Exhaust Cam
50. Air Cam



NATIONAL OIL ENGINE ELEVATION OF VAPORISER END.

DESCRIPTION OF PARTS—(*continued*).

51 Oil and Water Pump Cam
52. Oil Pump Lever Roller.
53. Cylinder S S Bracket.
54. Air Lever Fork
55. Exhaust Lever
56. Exhaust Lever Adjusting Screw.
57. Exhaust Shackle.
58. Exhaust Spring
59 Exhaust Spring Holder.
60. Exhaust Spring Holder Stud
61. Oil Pump Bracket.
62. Lamp Bracket
63. Lamp Bracket Stud.
64. Lamp Bracket Adjusting Screw.
65 Oil Pump Lever Fulcrum.
66. Oil Pump Valve Cover.
67 Oil Pump Spring
68 Water Injection Pump.
69. Governor Stop
70. Delivery Pipe from Water Pump
71. Governor Lever Fulcrum Pin
72. Air Lever Fork Pin.
73 Air Lever Fulcrum
74. Air Inlet Valve Box
75. Pipe connecting Oil Pump to Tank
76. Oil Pump Plunger Guide.
78. Inlet Water Flange
79 Exhaust Flange.
80 Air Flange
81 Drain Tap for Cylinder S S Bracket.
82 Air Pipe
84. Exhaust Spring Holder Set Screw
85. Pipe connecting Oil Pump to Spray
86. Exhaust Lever Fulcrum.
87. Governor Ball Pin
88 Governor Hanging Link Eye
89 Governor Fork
90. Governor Spindle.
97 Stud for Oil Tray.
98 Oil Tray.
99 Drain Cock for Oil Tray

Many slow-running light oil engines rely entirely upon the temperature of the vaporiser for the timing of the ignition, which, therefore, becomes rather precarious; in the National engine the impulses are timed by the action of the piston at the end of its stroke, working in conjunction with a bye-pass and ignition tube.

Cylinder and Valves.—The cylinder is a short casting, and contains the exhaust and main inlet valves. It is fitted with a separate liner of very hard cast iron, which is capable of easy replacement. The cylinder and liner are built into the frame of the engine, so as to secure rigidity, and an expansion joint is provided at the front of the liner, which is thus free to expand under the working heat.

The arrangement of the valves in the cylinder casting keeps them well away from the hot vaporiser; the cams on the side shaft for operating the valves are provided with large wearing surfaces, and are shaped with special regard to quiet action of the valves.

Instructions for Working.—The following extracts from the instructions for working issued by the makers will enable students to understand more fully the construction and operation of the engine.

Use of Heating-up Lamp—*Filling.*—Unscrew the cap (4) and fill the reservoir three parts full of petroleum, which should be passed through a strainer; replace the cap (4) and screw it down tightly.

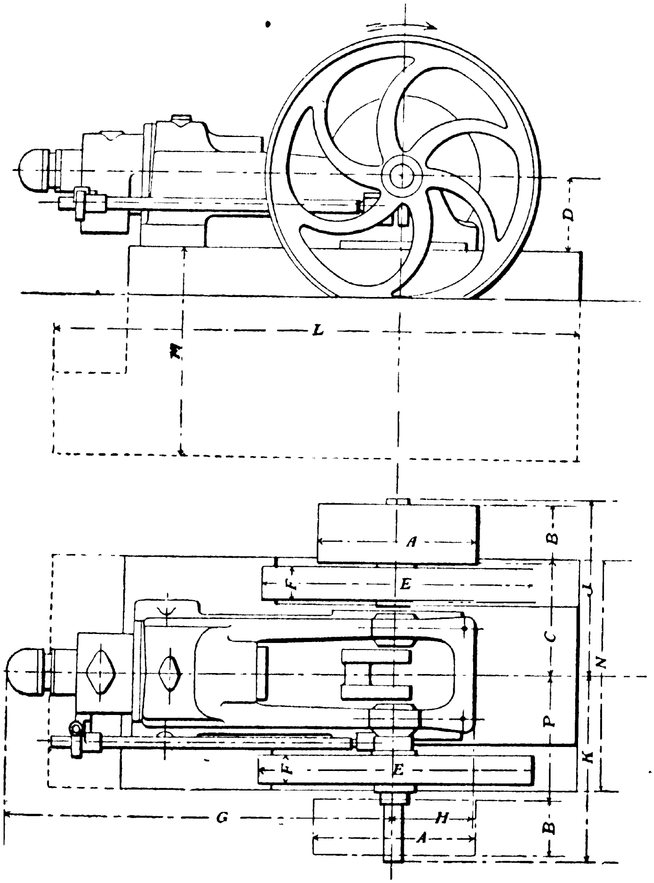
Lighting.—Open the air relief cock (5). Fill the heating-up dish (6) with methylated spirits or petrol and light it. When this has burned for two or three minutes, close the air-relief cock (5), and give a few strokes to the air pump (7). The lamp will now start working, the gases issuing from the nipple (8) being ignited by the flame from the spirit. When the spirit has burnt out, the burner will be sufficiently hot, and the lamp can be pumped up to full working.

Extinction.—To stop the lamp from working, unscrew the air relief cock (5) and leave it open.

Starting-up by Hand.—While the lamp is heating-up the vaporiser carry out the following operations :—

(1) Fill the lubricators and make sure that all the oil feed wicks are placed in the small tubes provided for them.

(2) Set the glass sight-feed lubricator to work by raising its spindle. Adjust the brass milled nuts so as to give the correct



number of drops per minute; this number varies from 8 drops in the smallest size to 24 in the largest.

(3) Move the roller (11) on the exhaust lever to the right, so

Power. Maximum B.H.P.	Speed in Revolutions per Minute.	PRINCIPAL OVERALL DIMENSIONS.																									
		Belt Pulley					Flywheel												Engine only.				Concrete Foundation.					
		Dia	Wid				Dia		Wid										Length.		Width.		Length.		Depth		Width	
		A	B	C		D	E		F	G		H	J		K		P		G + H		J + K		L		M		N	
		In.	In	Ft	In	In	Ft	In	In.	Ft	In	In	Ft	In	Ft	In	Ft	In	Ft	In	Ft	In	Ft	In.	Ft	In	Ft	In
2·25	320	8	4	1	1⅛	8½	2	6⅝	3 13/16	3	5	8½	1	6 7/16	1	8 7/16	1	2¾	4	1½	3	2⅞	5	0	2	9	2	0
3·25	320	10	5	1	1⅛	9½	2	6⅝	3 13/16	3	10	9½	1	6 7/16	1	8 7/16	1	2¾	4	7½	3	2⅞	5	0	2	9	2	0
4·5	300	12	6	1	1¼	10½	2	11½	4	4	0	11	1	8½	1	9⅝	1	3	4	11	3	6⅛	5	3	2	8	2	0
7·25	280	18	6	1	4½	12	3	6¾	4¾	4	11	12	1	11	2	1½	1	6	5	11	4	0½	6	6	3	1½	2	6
9	280	20	7	1	4	12	3	6¾	4¾	4	11	12	1	11	2	1½	1	6	5	11	4	0½	6	6	3	1½	2	6
10	260	24	10	1	5¾	13½	3	10	5¼	5	9	14	2	4½	2	6	1	8	6	11	4	10½	7	9	3	7½	3	0
13	250	24	10	1	6½	14	4	4¾	5	6	4	15	2	5½	2	6	1	9	7	7	4	11½	8	3	3	9	3	6
17·25	250	30	12	1	9¾	15	4	5½	6	7	1	16	2	11	3	2	2	1	8	5	6	1	9	0	4	3	4	0
22·5	250	30	12	1	10½	16	5	1¾	7¼	7	5	17	2	11	3	2	2	1½	8	10	6	1	9	6	4	7	4	0
27·5	240	40	12	2	3	17	5	3½	7¼	8	0	18½	3	5	3	8	2	6½	9	6½	7	1	10	6	4	6	4	8
33	230	40	14	2	3	18	5	8⅝	7¾	8	6	19½	3	6½	3	10	2	6½	10	1½	7	4½	11	0	4	9	4	8
40	230	54	14	2	4	18	5	11	7¾	8	8	19½	3	7	3	10½	2	7	10	3½	7	5½	11	0	4	9	5	0
47·5	230	54	16	2	5½	19	5	11	7¾	9	1	21	3	10½	4	2½	2	9½	10	10	8	1	11	6	5	0	5	3

Standard National Oil Engines fitted with two Flywheels.

as to bring it into contact with the relief cam; a steel pin (12) is supplied for keeping this roller in position.

(4) Turn the flywheel until the engine is in the correct position for operating the oil pump. The roller (52) on the oil pump lever (17) will then be against the flat on the oil cam (51).

During the time (five to seven minutes) necessary for these preparations the vaporiser will have become sufficiently heated to enable the engine to be started.

With the small handle (16) at the end of the oil pump lever (17), give a few smart downward strokes to the oil pump. To make this possible, the governor die (18) must be in such position that the pusher blade (19) will pass over it.

Now turn the flywheels quickly in the direction of rotation until the engine receives an impulse. As soon as the engine has received three or four impulses move the exhaust roller (11) to the left, back into its working position, and secure it there with the steel pin (12).

Water Injection.—This should be used only when the engine is working at three-quarters to full power, and the engine ought to run at least an hour before the water injection is necessary.

As soon as the vaporiser becomes too hot, and when the explosions appear to be violent, open the water cock (20). It is best so to regulate it that the explosions can just be heard. Care should be taken to see that the cock is closed when the engine is stopped, as otherwise the vaporiser would become flooded with water.

Stopping the Engine.—Push the oil pump lever (17) down as far as possible and insert the small brass wedge (44) (which is secured by a chain near the pump) into the space left between the small collar on the pump spindle and pump bracket.

After stopping the engine, turn the flywheels so that the crank is just above the "in-centre" on the firing stroke. In this position the air and exhaust rollers (11) and (46) will revolve freely. This ensures the closing of the valves, which is very important; inattention to this point may occasion trouble when re-starting.

LECTURE XIII.

DIESEL AND LIKE HEAVY OIL ENGINES.

The Essential Features of the Diesel Engine.—We have seen in Chapter IV. that the Diesel cycle is practically the same as the ideal constant pressure cycle; the Diesel engine was introduced by R. Diesel in about 1895, and very considerable interest was taken in it at the time, because the thermal efficiency measured upon the indicated horse-power was considerably in excess of anything which had been obtained up to that time. This increased efficiency was commonly stated to be due to the fact that the engine approached more nearly to the thermodynamically ideal engine than any other heat engine.

The essential feature of the Diesel engine is that no fuel is drawn in at the suction stroke; the air is compressed very highly on the compression stroke, the temperature being sufficient to cause to burn the oil which is pumped into the cylinder at the early portion of the working stroke. The result is that during the first part of the working stroke, while the oil is entering the cylinder, the pressure is kept nearly constant by the heat which continues to be generated. The first part of the working stroke, therefore, resembles that up to the cut-off in a steam engine. The compression ratio is very high, which makes for efficiency, and all risk of premature ignition which would occur with high compression in the ordinary internal-combustion engine is avoided, because the fuel only enters the cylinder when it is desired that combustion should occur.

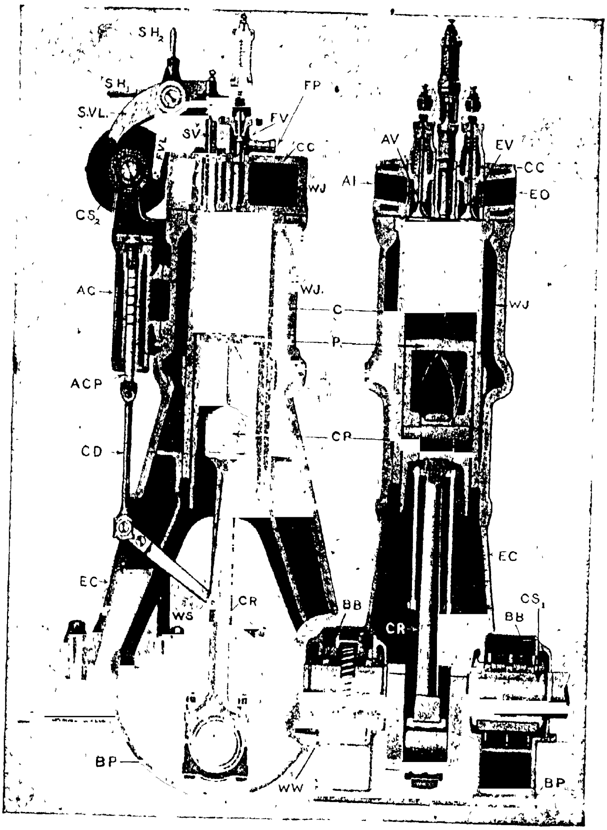
The very considerable success which the Diesel engine has achieved in practice is probably due more to the fact that it can be run on cheap crude or heavy oil than on account of its high thermodynamic efficiency.

In a test of a 500 H.P. Diesel engine made by Mr. M. Longridge, the I.H.P. was 595, after allowing for the negative work done in the pump by which the oil fuel was forced in. The consumption of oil was 207·2 lbs. per hour, or, roughly, ·35 lb. per I.H.P.-hour. Taking the calorific value in Centigrade units as 10,000 units per pound on the lower scale, these figures correspond to a thermal efficiency of 41 per cent., which is roughly twice as high as has been found in any steam engine, and is considerably higher than for any gas or light oil engine. In these tests, however, it has been calculated that if the pumps supplying oil and air at high pressure had been driven by the engine itself, the mechanical efficiency should be taken as ·77, so that the thermal efficiency reduces to 32 per cent.

This is an example of a point which we made in the first chapter of this volume; for practical purposes we must measure the thermal efficiency from the brake horse-power, because in some engines of high efficiency, such as the Diesel, considerable power is necessarily absorbed in the engine itself, so that the thermal efficiency based upon the indicated horse-power is a fictitious and rather misleading figure. We must also bear in mind that the indicated horse-power is calculated from an indicator card, the accuracy of which is not always free from suspicion.

The Diesel Oil Engine.

General Description and Method of Starting the Engine.—The Diesel Oil Engine is single acting and the details will be easily followed by comparing the index to parts with the one outside and two sectional views. The engine crank when placed just past the top centre by means of the barring handle BH, and if the starting handle be placed in position SH_1; then, compressed air is admitted from the starting air reservoirs through the starting valve SV to the cylinder C. This air at high pressure acts upon the piston P and causes the piston to move down. Further, during the upstroke of the piston the previously admitted air exhausts from the cylinder C through the exhaust valve EV and outlet EO to the atmosphere. After the engine has made 5 or 6 revolutions the starting handle SH, is moved into the vertical position SH_2; thus cutting off the air supply by throwing the lever and roller SVL out of gear, and at the same time bringing the fuel valve lever FVL and fuel valve FV into action. Now, the high pressure air from the blast receiver (not shown in the illustrations), which passes round the fuel valve FV carries oil as a fine spray from the fuel valve casing into the cylinder immediately the fuel valve FV is opened. When the oil spray is ignited by the heat produced during the previous compression of the air inside the cylinder, the engine is ready for taking a load.



CROSS AND LONGITUDINAL VERTICAL SECTIONS OF A DIESEL OIL ENGINE.

(See the facing page for the Index to Parts, and the third page after this for the outside view)

Index to Parts.

BH	represents	Barring Handle to set the piston at beginning of its down stroke (*see the outside view*).
SH_1	,,	Starting Handle in position 1.
CS_2	,,	Cam Shaft and cams.
SVL	,,	Starting Valve Lever and roller.
SV	,,	Starting Valve.
AI	,,	Air Inlet Pipe for admitting air from the starting receivers at pressure of 800 to 900 lbs. per square inch.
AV	,,	Air Valve to cylinder.
EV	,,	Exhaust Valve.
EO	,,	Exhaust Outlet pipe.
CC	,,	Cylinder Cover.
WJ	,,	Water Jacket.
C	,,	Cylinder.
P	,,	Piston.
CP	,,	Crosshead Pin.
CR	,,	Connecting-Rod.
CS_1	,,	Crank Shaft.
BB	,,	Bearing Bushes.
BP	,,	Bed-Plate.
EC	,,	Engine Column.
SH_2	,,	Starting Handle in position 2, with SVL and starting valve thrown out of action.
FVL	,,	Fuel Valve Lever and roller.
FV	,,	Fuel Valve.
FP	,,	Fuel Pipe from oil pumps OP, and air blast supply from blast receiver at a pressure of 800 lbs. per square inch.

Air Compressor Gear.

CD	,,	Compressor Driving Gear from CR.
ACP	,,	Air Compressor Piston.
AC	,,	Air Compressor cylinder.
WAC	,,	Water Air-Cooler (*see outside view*).

Governor and Cam-Shaft Gear.

WW	,,	Worm Wheel on CS_1 for driving Governor and cam shaft.
WS	,,	Worm Shaft (*see outside view*).
G	,,	Governor (*see outside view*).

The Working Cycle.—This engine operates upon the well-known four-stroke Otto cycle

1. On the first actual working down stroke of the piston, air is drawn direct from the atmosphere into the cylinder C through the air inlet AI and air-valve AV.

2. On the first upstroke this air is compressed, which raises its temperature sufficiently to ignite the fuel, when this is admitted to the cylinder as indicated above.

3. At the point of maximum compression, and during the first period of the next down stroke, an oil spray is blown into the cylinder by compressed air from the blast receiver, through the fuel pipe FP and fuel valve FV.

4. On the second upstroke the products of combustion are expelled to the atmosphere past the exhaust valve EV and through the exhaust outlet pipe EO.

Advantages.—The engine is a vertical one and the four valves, SV, AV, FV, EV are arranged in the cylinder cover. Each valve has a separate

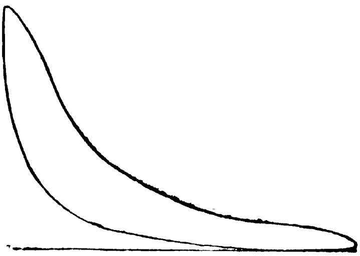


DIAGRAM TAKEN WITH A "M'INNES-DOBBIE" EXPLOSIVE ENGINE INDICATOR, OF THE SMALL SIZE DESIGN WITH ONE EXTERNAL SPRING, FROM A DIESEL OIL ENGINE. INITIAL PRESSURE, 520 LBS PER SQ INCH; SPRING, $\frac{1}{240}$ TO THE INCH; REVS. PER MINUTE, 202; B.H.P 80.

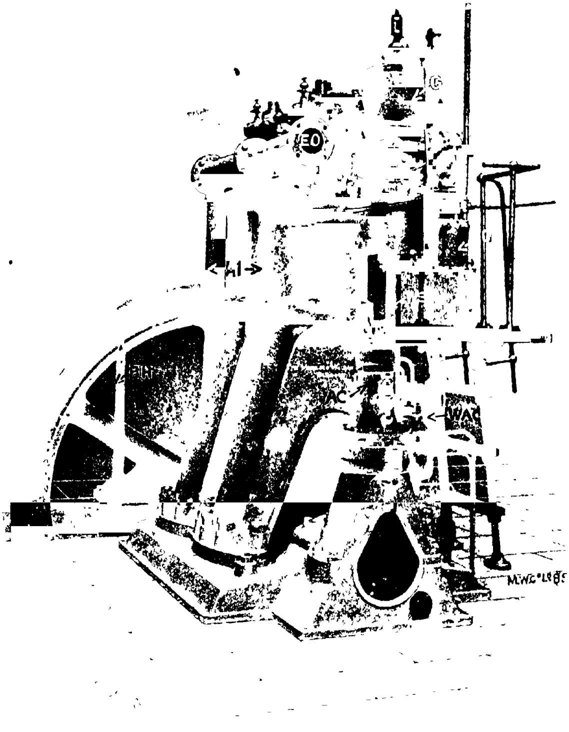
seat, which can be removed without displacing the valve-driving gear or lifting the cylinder cover.

These valves are mechanically opened by cams upon the shaft CS_2. This shaft is driven from the wormwheel WW on the crank shaft, through the worm shaft WS, at half the speed of the engine crank shaft CS_1

The compressed air which is required for starting the engine and for blowing the fuel into the cylinder is supplied by the air compressor AC.

In the latest form of the Diesel Oil Engine the air is compressed in two stages and the compressor is directly coupled to the end of the crank-shaft CS_1, as shown by the outside view.

It will be seen from the accompanying indicator diagram that there is no explosion or sudden rise of pressure in the cylinder C, hence great smoothness in running.

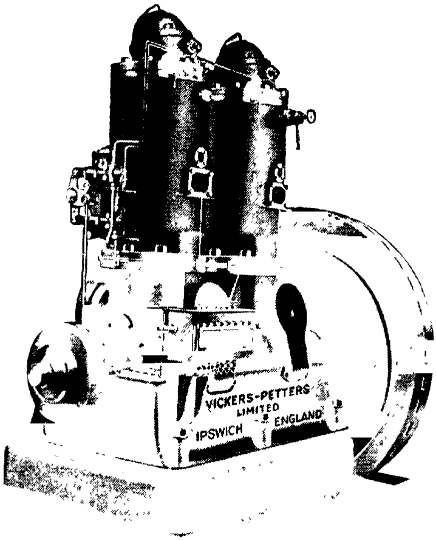


OUTSIDE VIEW OF A DIESEL OIL ENGINE.

MADE BY MIRRLEES, WATSON & CO., GLASGOW

The Vickers-Petter Semi-Diesel Engine.—The term semi-Diesel is employed to describe an engine which burns crude oil but does not carry the compression high enough to cause combustion of the fuel.

The Vickers-Petter engine works upon the two-stroke cycle, with crank-chamber compression of the air and the admission of the compressed air to the cylinder is by way of inlet ports which are over-run by the piston. The exhaust ports are on the opposite side of the cylinder barrels to the inlet ports, and are over-run by the piston when near the bottom of the stroke. At the moment of maximum compression fuel is injected into the



VICKERS-PETTER SEMI-DIESEL ENGINE.

hot-bulb combustion chamber through a sprayer by means of an oil pump, the variable stroke of which is controlled by a governor.

The following table gives the leading dimensions and other useful data of the various sizes of this engine.

B.H.P. (Working load).	Revolutions per Minute.	Number of Cylinders	Leading Dimensions						
			Diameter of Piston.	Length of Stroke	Diameter of Flywheel	Width of Flywheel	Overall Height	Overall Width.	Overall Length.
10	425	1	7¼″	8″	3′ 6″	4¾″	3′ 10″	3′ 6″	4′ 9″
13½	400	1	8¼″	9″	4′ 0″	6″	4′ 5″	4′ 0″	5′ 1″
18	350	1	9¾″	10½″	4′ 6″	6″	5′ 3″	4′ 6″	5′ 7″
25	300	1	10¾″	11½″	5′ 0″	8″	5′ 8″	5′ 0″	6′ 0″
35	275	1	12″	14″	6′ 0″	8″	6′ 6″	6′ 0″	7′ 3″
50	260	1	14″	16″	6′ 6″	10″	7′ 6″	6′ 6″	8′ 0″
75	250	1	16″	18″	7′ 0″	12″	8′ 8″	8′ 0″	9′ 9″
70	275	2	12″	14″	5′ 6″	10″	6′ 9″	5′ 0″	10′ 6″
100	260	2	14″	16″	6′ 0″	12″	7′ 8″	6′ 0″	11′ 8″
150	250	2	16″	18″	6′ 6″	12″	8′ 8″	8′ 0″	13′ 0″
200	260	4	14″	16″	6′ 0″	12″	7′ 0″	6′ 0″	13′ 7″
300	250	4	16″	18″	6′ 6″	12″	8′ 8″	8′ 0″	20′ 0″
450	250	6	16″	18″	6′ 6″	12″	8′ 8″	8′ 0″	27′ 6″

Test Results.—The following test results give a good idea of the fuel consumption and "flexibility" of this type of engine:—

(*a*) 35 B.H.P. Engine at 275 Revolutions per Minute run on Admiralty Fuel Oil supplied by the British Petroleum Company. Sp. Gr. = ·930 at 68° F.; Calorific Value, 19,300 B.Th.U. per Lb.

	Normal Load Test	Overload Test at accelerated speed	Reduced Speed Test
Duration of test,	2 hours	1 hour	2 hours
Revolutions per minute,	256	300	215
Actual B.H.P. developed,	34·7	46·7	29·1
Total fuel consumed,	28 pints	19 pints	25 pints
Fuel consumed per B.H.P.-hour,	·404 pint	·406 pint	·43 pint

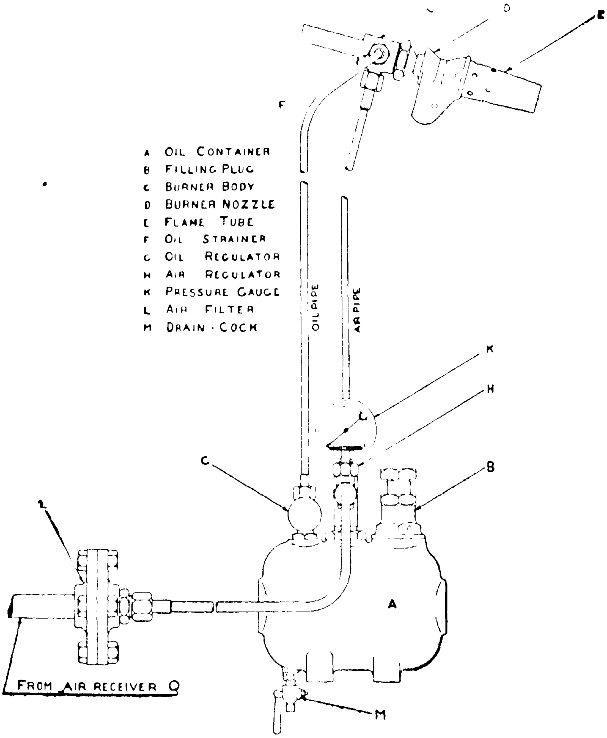
(*b*) 50 B.H.P. Engine at 250 Revolutions per Minute run on Scotch Shale Oil.

	Normal Load Test	Overload Test at Normal Speed.	Reduced Speed Tests	Slow Speed Test.
Duration of test,	6 hours	1 hour	3 hours	2 hours
Revolutions per minute,	250	252	214	178
Actual B.H.P developed,	50·7	55·7	42·0	35 0
Total fuel consumed,	132 pints	24 pints	57·5 pints	32·7 pints
Fuel consumed per B.H.P.-hour,	·433 pint	·431 pint	·455 pint	·467 pint

(*c*) 200 B.H.P. Engine at 250 Revolutions per Minute run on Gas Oil, ·895 Sp. Gr.

Duration of test,	5 hours.
Revolutions per minute,	300
Actual B.H.P. developed,	232
Total fuel consumed,	472 pints.
Fuel consumed per B.H.P.-hour,	·407 pint.

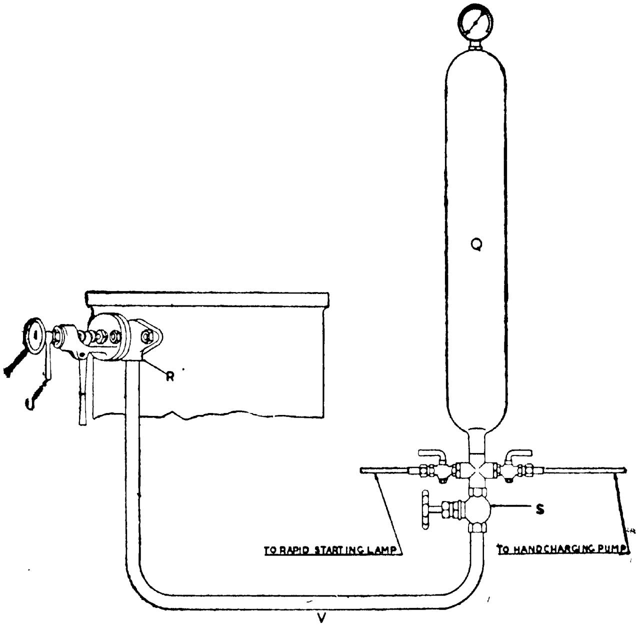
Starting Burner.—For starting the engine a special rapid starting liquid fuel burner is provided; this burner is operated by compressed air or inert gas and requires no pre-heating. The oil container A is filled with paraffin, and a wick is dipped



STARTING BURNER FOR VICKERS-PETTER SEMI-DIESEL ENGINE.

in paraffin and placed in the pocket E[1]; the wick is then ignited, care being taken to see that the flame extends inside the tube E. The compressed air is then turned on by opening the valve

shown below, and the oil regulator G is opened; the air regulator is then opened carefully until a pressure of 20 to 25 lbs. per square inch is shown on the pressure gauge K. The oil is thus forced to the burner by the pressure of air in the container, and is then ignited by the wick, the proportion of air and oil leaving the container being carefully regulated to give the desired intensity of flame; an excess of oil to the burner will



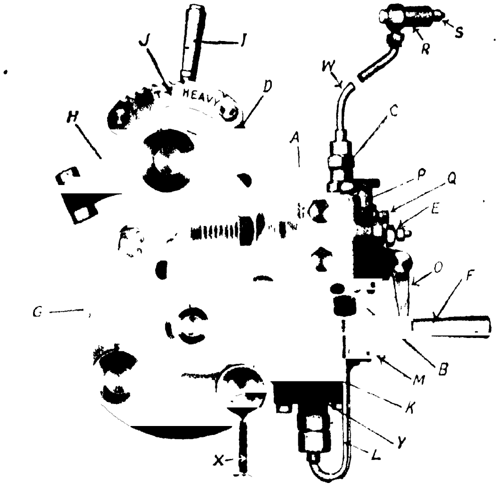
COMPRESSED AIR STARTER, VICKERS-PETTER SEMI-DIESEL ENGINE.

be indicated by a yellow flame, while an insufficiency of oil will be indicated by a short, fierce flame.

Compressed Air Starting Gear.—The compressed air starting apparatus consists of a valve R and a pressure receiver Q, into which compressed air or inert gas is forced and stored during the working stroke of the engine, when next it is required to

start. When connecting the pipes, care must be taken to make all joints air-tight and to ensure that the pressure gauge is screwed in firmly. When starting the larger engines for the first time, the receiver is charged by means of the hand pump supplied.

When the engine is running, the pressure receiver may be charged in the following manner:—Open the valve S and turn hand wheel T to allow the starting valve to act as a check or back pressure valve, and the receiver will be charged to about 150 170 lbs. per square inch in a few minutes. Do not allow



LIGHT-RUNNING GEAR, VICKERS-PETTER SEMI-DIESEL ENGINE.

A, Fuel Pump Body.
B, Suction Valve.
C, Delivery Valve.
D, Pump Plunger.
E, Safety Diaphragm Plug.
F, Fuel Pump Handle.
G, Rocker Arm.
H, Auxiliary Rocker Arm.
I, Hand Operating Lever.
J, Notched Quadrant.
K, Equalising Chamber.
L, Fuel Pump Suction Pipe.
M, Fuel Feed Pipe Connection
O, Pump Handle Stop.
P, Pump Handle Adjusting Screw.
Q, Equaliser Air Release Cock.
R, Sprayer Body.
S, Sprayer Nozzle.
W, Delivery Pipe.
X, Eccentric Rod.
Y, Connector carrying Strainer

the starting valve too much lift or it will become overheated. After the receiver is charged, close the stop valve S and with the hand wheel T close the starting valve R.

Light-Running Gear.—The light-load gear for single-cylinder engines consists of an auxiliary rocker arm H mounted on an eccentric, which, when in use, operates the fuel pump 180° in advance of the normal timing, by depressing the fuel pump hand lever F.

To operate, move the handle I over towards the light-load mark until a slight stroke is given to the fuel pump hand lever F, the stroke being regulated to suit the load on the engine and to maintain sufficient heat in the combustion chamber to ensure regular firing and steady running.

The handle I should not be moved into the light-load position when the engine is stopped, unless the fuel pump handle F is depressed.

The Diesel Engine for Marine Propulsion.—Considerable progress has been made in recent years in the application of oil engines to marine propulsion, so that the "oiler" is gradually becoming a serious rival to the "steamer."

We are indebted to the author, and to the Secretary of the Institute of Engineers and Shipbuilders of Scotland for permission to give the following quotations from a paper read before that body by Mr. James Richardson, B.Sc., on "The Present Position (1920) of the Diesel Engine for Marine Propulsion":—

The present position reflects clearly the finding of past performance. Eighty-four per cent. of the ships are twin-screw, and over 80 per cent. of the total number of marine engines and horse-power is of the four-stroke cycle type. In Britain the two-cycle is more favoured than elsewhere.

The most important of the problems of design are common to both two and four-cycle engines, and the first concerns the injection of the fuel into the working cylinder. The exact quantity of fuel, at a pressure sufficient to ensure injection, must be measured out by the pump and spread in some 30° of revolution as widely as possible into the combustion space, in a sufficiently finely-divided state to ensure rapid ignition and satisfactory combustion. Chiefly in this direction of improving distribution of the fuel in the combustion chamber can increased economy be sought.

There are two alternative methods of spraying; one by means of compressed air, the other by injecting the fuel at a high

pressure, known as the solid-injection system. The utilisation of compressed air is most general. So far as marine installations are concerned, the principal advantage of the solid-injection method, where compressed air is not used to assist injection, and is only required for supplying manœuvring power, is that the compressors are not a part of the main propelling engine, and do not require to run continuously at sea. Moreover, the air-compressing plant, where solid injection is adopted, can be reduced by from 40 per cent. to 50 per cent. in capacity.

Reliability with air compressors delivering at 850 lbs. to 1,000 lbs. per square inch has now reached a high level, the leading factors towards this end being generally appreciated, and air-compressor problems rated at their full value. Multi-staging is essential, and compression to 1,000 lbs. per square inch should be carried out in not less than three stages, so proportioned that no undue ratio of compression and consequent temperature can occur in any stage Particularly should the first or low-pressure compression be minimised Such rises of temperature as are inevitable with compression should be reduced by efficient cooling of the air during compression and by the installation of inter-coolers and after-coolers to reduce to atmospheric whatever temperature remains after compression. Removal of moisture from the air should be facilitated. Every effort in design should be made to ensure that the compressor valves and springs can be easily removed and replaced.

It is not to be inferred in any way that finality has been reached in compressor design or in methods of injection. In regard to the former, a better means for removing oil and moisture from the air is required, and in respect of injection a suitable valve should be produced which, whilst requiring compressed-air spray for full power and maximum efficiency, can still operate satisfactorily at reduced power without the medium or assistance of compressed air.

The other fundamental of injection, the measuring of the oil fuel, involves the most delicate apparatus associated with the Diesel oil engine—the fuel-injection pumps and controlling gear. The system of having one pump per cylinder is now almost universal for marine work (excepting where solid injection is used), and only in this way can the maximum security against a large percentage of overcharge and overload in one or more of the cylinders be obtained. These individual pumps, therefore, become of relatively small size. Even with separate pumps,

NOTABLE DIESEL SHIPS.

Date.	Name of Vessel	Dimensions	length. ft	beam ft	depth ft	Makers of Machinery.	Type of Engine.	
1910	Vulcanus,	1,900 dispt.	208	37¾	13	Werkspoor,	Werkspoor,	1
1912	Selandia,	10,000 dispt.	386	53	30	Burmeister and Wain,	Burmeister and Wain,	2
1912	Jutlandia,	10,000 dispt.	386	53	30	Barclay Curle,	Burmeister and Wain,	3
1912	Monte Penedo,	6,500 dispt.	350	50	27	Sulzer,	Sulzer,	4
1912	Juno,	4,300 dispt.	258	45	28	Werkspoor,	Werkspoor,	5
1913	Siam,	13,200 dispt	410	55	30½	Burmeister and Wain,	Burmeister and Wain,	6
1913	Wotan,	7,960 dw.	404	52½	29½	Reiherstieg,	Carels,	7
1913	Hagen and Loki.	7,900 dw.	400	53	32¼	Krupp,	Krupp,	8
1913	Arum,	5,600 dispt.	360	47	27	Swan Hunter,	Swan Hunter "Neptune" type.	9
1914	Fionia,	7,000 dw.	395	53	30	Burmeister and Wain,	Burmeister and Wain,	10
1916*	Abelia,	5,600 dispt.	360	47	27	Wallsend Slipway and Engineering Co.	Maschinen-Fabrik, etc. (M.A.N.).	11
1916	Trefoil,	4,510 dispt.	280	39	23½	Vickers,	Vickers,	12
1916	Hamlet,	10,050 dispt.	360	56	29¾	Atlas Diesel, Sweden,	Polar,	13
1918	Ansaldo San Giorgio I.	8,100 dw.	378	50⅔	30¼	Ansaldo San Giorgio,	F.I.A.T. (now Ansaldo San Giorgio).	14
1919	Glenapp,	19,000 dispt	450	55⅔	36½	Harland and Wolff,	Burmeister and Wain,	15
1920	Afrika,	18,600 dispt.	445	60	42	Burmeister and Wain,	Burmeister and Wain,	16
1920	Glenogle,	19,000 dispt.	502	62	27½	Harland and Wolff,	Burmeister and Wain,	17
1920	Fritz,	4,000 dw.	331	44⅔	—	Blohm and Voss,	M.A.N.,	18
1920	Maumee,	15,000 dispt.	455	56	26	New York Naval Yard,	M.A.N.,	19
1920	Cubore,	17,000 dispt.	450	57	37	Bethlehem Steel Co	—	20
1920	Zoppot,	22,000 dispt.	525¾	66½	33½	Krupp	Krupp,	21
1920	Salerno,	6,500 dw.	375	51	34	Werkspoor,	Werkspoor,	22
1920	Narragansett,	14,000 dispt.	425	56⅔	33	Vickers,	Vickers,	23
1920	Fullagar,	500 dw.	150	23¾	11½	Cammell Laird,	Fullagar,	24

	Single or Twin	Cycle.	I H.P per Engine.	B H P per Engine.	No of Cylinders per Engine	B.H P per Cylinder.	Diameter of Cylinders.	Stroke.	Ratio. Stroke/Bore	R P M.	Piston Speed.	M E P on B H P Basis.	M E P. on I H.P. Basis.
							in	in			ft per min		
1	Single	4 S.A.	490	390	6	65	$15\frac{3}{4}$	$23\frac{5}{8}$	1·5	140	550	79·0	99·0
2	Twin	4 S.A.	1,250	1,050	8	130	$20\frac{7}{8}$	$28\frac{3}{4}$	1 38	140	670	75·0	89·8
3	Twin	4 S.A.	1,250	1,050	8	130	$20\frac{7}{8}$	$28\frac{3}{4}$	1·38	140	670	75·0	89·8
4	Twin	2 S.A.	1,200	850	4	212	$18\frac{1}{2}$	$26\frac{3}{4}$	1·44	160	715	73·0	103·0
5	Single	4 S.A.	1,460	1,100	6	183	$21\frac{1}{8}$	$39\frac{3}{8}$	1·87	125	820	84·0	111·0
6	Twin	4 S.A.	1,580	1,350	8	169	$23\frac{5}{8}$	$31\frac{5}{8}$	1·34	125	660	77·0	90·0
7	Single	2 S.A.	2,100	1,650	6	275	$23\frac{5}{8}$	$43\frac{1}{4}$	1·83	90	650	64·0	81·5
8	Twin	2 S.A.	1,600	1,200	6	200	$18\frac{7}{8}$	$31\frac{7}{8}$	1·69	135	720	66·0	88·0
9	Twin	2 S.A.	850	600	4	150	$17\frac{1}{4}$*	33	1·91	120	660	64 0	90·5
10	Twin	4 S.A.	2,140	1,600	6	266	$29\frac{1}{8}$	$43\frac{1}{4}$	1 49	100	720	74 0	99·0
11	Twin	2 S.A.	850	600	4	150	$17\frac{1}{4}$	33	1·91	120	660	64·0	90·5
12	Twin	4 S.A.	1,000	750	8	94	17	27	1·59	150	675	81·0	108·0
13	Twin	2 S.A.	2,300	1,650	6	275	24	36	1·5	120	720	67·0	93·5
14	Twin	2 S.A.	1,450	1,100	4	275	$24\frac{3}{4}$	$35\frac{1}{2}$	1·43	100	590	64·0	84·5
15	Twin	4 S.A.	3,200	2,625	8	328	30	$43\frac{1}{4}$	1·44	115	830	74·0	90 5
16	Twin	4 S.A.	2,250	1,700*	6	283	$29\frac{1}{8}$	$45\frac{1}{4}$	1·55	115	865	65·0	86 0
17	Twin	4 S.A.	3,200	2,625	8	328	$29\frac{1}{8}$	$45\frac{1}{4}$	1·55	115	865	75·0	91·5
18	Twin	2 D.A.	1,250*	850	3	283	$18\frac{7}{8}$	28	1·48	110	515	65 0	95·5
19	Twin	2 S.A.	3,200	2,500	6	416	$25\frac{1}{8}$	$37\frac{3}{8}$	1·48	130	810	69·0	88·0
20	Single	2 S.A.	3,900	2,700	6	450	$25\frac{1}{2}$	48	1·87	100	800	73 0	105 0
21	Twin	2 S.A.	2,000	1,400	6	233	$22\frac{5}{8}$	$39\frac{3}{8}$	1·73	106	695	55·0	79 0
22	Twin	4 S.A.	1,400	1,050	6	175	22	$39\frac{3}{8}$	1·78	125	820	74·0	98·5
23	Twin	4 S.A.	1,620	1,250	6	208	$24\frac{1}{2}$	39	1·59	118	767	76 0	99·0
24	Single	2 O.P.	660*	500	4	125	14	20×2	1·43	110	367	73·5	92·0

continual care and intelligent supervision must be exercised to ensure that the deliveries from the pumps are maintained equal, lest one or more cylinders should be overloaded to counteract the effect of diminished deliveries from defectively working pumps. Means are now generally provided whereby any pump can be cut out and overhauled whilst the remainder are in operation.

The Diesel engine remains a heavy and expensive prime mover in comparison with its steam rivals. Even with the progress made in design within the last six years, it can definitely be stated that there has been generally an increase in the weight and the space occupied by the slow-speed Diesel engine per horse-power developed continuously. The factors opposed to a reduction of these disabilities and rendering difficult the path toward the higher powers now desired, are the temperature gradient through the metal surrounding the combustion chamber and the fact that the major portion of the material of the engine is only utilised for a small fraction of the running time. The former refers particularly to the two-cycle, and the latter to the four-cycle engine, where three-quarters of the running time is idle so far as power output is concerned. Moreover, of the one power stroke per two revolutions, only one-half of this stroke, or 12·5 per cent. of the cycle, stresses the parts of the engine camparably with their strength and scantlings.

The unique economies possible by the adoption of the principle of internal combustion are only gained at the expense of foregoing all the many advantages derived from using the most flexible known power-conveying medium, steam.

To appreciate the type of stressing to which the main parts of a Diesel engine are subject comparison may be made with a steam engine of normal design and of equal power output. The maximum normal load with the oil engine is more than five times as great, and, furthermore, the rate of application of this main load is increased 10 times.* With internal combustion the normal stresses are liable to sudden increases calling for a larger factor of safety, and design questions relating to the main parts of the engine must be considered strictly in this light.

In regard to the temperature gradient and the stresses so incurred, data are lacking; but, so far as they are available, they go definitely to prove that the four-stroke cycle engine

* *Proc. of the Junior Institution of Engineers*, May, 1914 "High Power Diesel Engines: Their Development for Marine Service."

has distinct advantages. It can be shown that the temperature gradient through the metal of a four-cycle cylinder liner and through the furnace of a cylindrical marine boiler are substantially equal. In the former the stressing is intermittent, and in the latter relatively steady. The stresses consequent upon this temperature gradient are half as high again in the case of the boiler furnace, due to the higher coefficient of expansion and modulus of elasticity of steel as compared with cast-iron. With the cylinder liner the transmission of heat is a subsidiary function to the more important duties of guiding the power piston and maintaining a suitable surface to permit piston-ring gas-tightness with minimum friction. The simple design of liner adopted up to the present with all but a few exceptions cannot be approved on theoretical grounds To obtain a minimum value for the combined stress consequent upon temperature gradient through the metal and internal pressure, a ribbed and built-up form of construction must be adopted. This can very simply be accomplished, as is sometimes the case, with the four-cycle engine, but imposes greater difficulties in the case of the two-cycle, where exhaust and scavenging ports have to be accommodated in the cylinder. Computation of the high stresses with the two-cycle cylinder in way of scavenging air-inlet and main exhaust ports is extremely difficult. These are bound to be very considerable in view of the high velocity of the exhaust gases and the consequent high rate of heat transmission from these gases to the walls, and are further intensified by cooling and distorting effect of the entering scavenging air with the normal two-cycle design

At present the cost can be stated to be from 25 per cent. to 33 per cent more than for a steam plant, depending on the type of auxiliaries applied to the oil-engined ship, and whether comparison is made with reciprocating or double-reduction turbine steam machinery. This higher cost is minimised in effect by the greater cargo-carrying capacity sometimes possible by the adoption of oil engines, or alternatively by the fact that a smaller Diesel-engined ship serves to give earning capacity equal to a steamer. The economy of operation possible with the oil engine is such that, granted reliable performance, the extra cost is speedily balanced by the increased profits obtainable. The total saving for 200 days' sailing per annum by installing Diesel instead of steam machinery is given for wages and keep, fuel and oil costs, in the case of a single-screw ship of 1,000 brake horse-power,

COMPARISON OF RUNNING COSTS OF DIESEL SHIPS AND STEAM SHIPS OF 1,000 BRAKE HORSE-POWER.

	Single-Screw, Diesel, 1,000 B H P.	Single-Screw, Double-Reduction Geared, 1,000 S.H P.		Single-Screw, Reciprocating, 1,200 I H.P.	
		Coal.	Oil.	Coal.	Oil
Fuel, pounds per horse-power per hour,	0·45	1·5	1·1	1·95	1·4
Consumption, tons per day,	4·82	16·1	11·8	25·0	18·0
Consumption, tons per 30 days,	145	483	354	750	540
Price of fuel per ton,	£11	£5	£10	£5	£10
Cost of fuel per 30 days,	£1,595	£2,415	£3,540	£3,750	£5,400
Lubricating oil consumption, gallons per day,	10	2	2	3	3
Lubricating oil, cost per gallon,	5s.	5s.	5s.	5s.	5s.
Lubricating oil, cost per 30 days,	£75	£15	£15	£22 10s.	£22 10s.
PERSONNEL—					
Chief engineer,	1	1	1	1	1
Assistant engineers,	3	2	2	2	2
Greasers,	3	3	3	3	3
Firemen,	..	3	3	3	3
Trimmers,	..	3	..	3	..
Donkeyman,	1	1	1	1	1
Electrician,	1	..	..	..	..
Total engine-room staff,	9	13	10	13	10
Total wages, 30 days,	£191	£224 10s.	£195	£224 10s.	£195
Total keep per 30 days, at 7s. per day,	£94 10s.	£136 10s.	£105	£136 10s.	£105
Total wages, fuel, oil, and keep for 30 days,	£1,955 10s.	£2,791	£3,855	£4,135 10s.	£5,722 10s.
Ratio,	1	1·44	1·97	2·12	2·93
Net saving per annum of 200 days' sailing, Diesel over steam.	.	£5,570	£12,650	£14,520	£25,250

Note.—In addition to the above, the following savings are effected :—Fuelling costs, less demurrage, additional cargo capacity, less accommodation for engine-room staff, no stand-by losses, less cleaning ship, higher average speed in a sea-way, reduced fueling appliances required, etc.

Cost of Fuel Oil and Coal at Principal Ports, July, 1920.

Port.	Fuel Oil per Ton	Coal per Ton
	s. d.	s. d.
Alexandria,	250 0	186 0 to 200s.
Adelaide,	180 0	40 0
Batavia,	150 0	127 0
Bombay,	150 0	45 0
Buenos Aires,	265 0	170 0
California,	62 0 to 92 0	69 0
Christiania,	224 0	210 0
Calcutta,	250 0	25 0
Cape Town,	220 0	46 9 Transvaal.
Colombo,	150 0	102 6
Curacao,	80 0	125 0
Glasgow,	250 0	115 0 Welsh.
Hong Kong,	150 0	115 0 Welsh.
Havana,	142 6	125 0
Karachi,	150 0	45 0
London,	250 0	115 0 Welsh.
Liverpool,	250 0	115 0 Welsh.
Lisbon,	250 0	160 0
Madras,	150 0	45 0
Melbourne,	180 0	35 0
New Orleans,	58 2	40 0 to 69s.
New York,	47 6	55 0 to 68s.
Palembang,	125 0	127 0
Pensacola,	80 0	42 0
Port Said,	250 0	186 6
Panama,	75 0	125 0
Rotterdam,	220 0	160 0
Rio de Janeiro,	250 0	185 0
St. Thomas,	160 0	124 0
Sydney,	175 0	21 3

Average price per ton :—Coal = 104s. ; fuel oil = 191s.

and also a ship of 2,400 brake horse-power; twin-screw in the case of Diesel engines and single-screw in the case of steam. No estimation has been made regarding the other savings generally possible, the most important of which is additional cargo capacity, no stand-by losses, and higher average speed, more particularly when the oil is compared with the coal-fired ships. The figures, however, for the savings in fuel, lubricating oil, wages and keep are, it will be admitted, sufficiently attractive to merit close attention by shipowners to the Diesel system of propulsion, when full comparisons can readily be made to meet the specific case of any particular type of ship or trade.

LECTURE XIV.

PRODUCER GAS PLANTS.

Theory of Producer Gas.—Producer gas is obtained by forcing or sucking air through a mass of highly heated fuel, with the result that the carbon is oxidised to carbon monoxide (CO). In order to utilise the high temperature produced, steam is admitted along with the air, the resulting hydrogen formed assisting materially to increase the calorific value of the resulting gas.

When the air and steam are forced through the fuel by pressure, the producer is called a *Pressure Producer;* and when they are drawn through by suction caused by the suction strokes of the engine, the producer is called a *Suction Producer.*

In general it may be stated that pressure producers are most suitable for high powers, suction producers being specially suitable for low powers.

The simple theory governing both forms of plant can be expressed by the following equations:—

$$C + O_2 = CO_2. \quad \ldots\ldots \quad (1)$$

$$CO_2 + C = 2CO. \quad \ldots\ldots \quad (2)$$

In these reactions the carbon in the fuel first burns to CO_2 at the bottom of the fire, and then becomes reduced to CO as it passes through the remainder of the heated fuel. It is probable that some of the carbon does become first converted to CO_2 and then reduced in this manner, and that some of the carbon monoxide is formed directly from the carbon in the manner indicated by the formula—

$$2C + O_2 = 2CO. \quad \ldots\ldots \quad (3)$$

The calorific value of carbon monoxide is about 10,200 B.Th.U. per pound, while that of carbon is about 14,500 B.Th.U. per pound, so that if no steam were used, the maximum possible efficiency of the plant would apparently be $\frac{10,200}{14,500}$ i.e., about 70 per cent.

Part of the 30 per cent. apparently lost is utilised in decomposing the steam; in this process the following two reactions both occur:—

$$C + H_2O = CO + H_2. \quad . \quad . \quad . \quad . \quad . \quad (4)$$
$$C + 2H_2O = CO_2 + 2H_2. \quad . \quad . \quad . \quad . \quad (5)$$

Reaction (4) takes place at temperatures above about 1,800° F., but at about 1,100° F. and under reaction (5) occurs. The gas obtained by equation (4) has a higher calorific value and the reaction absorbs more heat, so that the best results should be obtained in practice by working at the highest temperature possible under practical conditions.

Dowson's Analysis of Heat Reactions.—Mr. J. E. Dowson, who invented the first of these plants, gave the following analysis of these reactions:—

Taking weights equal to molecular in pounds—

$$C + O_2 = CO_2 + 176{,}000 \text{ B.Th.U.} \quad . \quad . \quad (1a)$$
$$2C + O_2 = 2CO + 106{,}000 \text{ B.Th.U.} \quad . \quad . \quad (3a)$$

The former gives a gas having a calorific value of about 119 B.Th.U. per cubic foot.

Corresponding to equations (4) and (5), we shall have—

$$C + H_2O = CO + H_2 - 51{,}800 \text{ B.Th.U.} \quad (4a)$$
$$C + 2H_2O = CO_2 + 2H_2 - 33{,}600 \text{ B.Th.U.} \quad (5a)$$

The minus sign indicates an absorption of heat.

If the steam is admitted to the producer in the form of water at about 60° F., approximately 1,120 B.Th.U. will be absorbed per pound of the steam, and this must be added to the figures given in equations (4a) and (5a).

Equation (3a) shows that with 24 lbs. of carbon burnt 106,000 B.Th.U. are liberated. Corresponding to 12 lbs. of carbon with equation (6a), we shall employ 36 lbs. of water, absorbing $36 \times 1{,}120 = 40{,}300$ B.Th.U., so that, in all, $40{,}300 + 33{,}600 = 73{,}900$ B.Th.U. are absorbed.

$\therefore$ Pounds of water required to absorb all the heat of reaction $(2a) = \dfrac{36 \times 106{,}000}{73{,}900} = 51 \cdot 6$.

$\therefore$ Total amount of carbon burnt $= 24 + \dfrac{51 \cdot 6 \times 12}{36} = 41 \cdot 2$ lbs.

$\therefore$ Pounds of water required per pound of carbon $= \dfrac{51 \cdot 6}{41 \cdot 2} = \mathbf{1 \cdot 25}$.

Theoretical Analysis and Calorific Value of Producer Gas.—From equation ($3a$) we see that 24 lbs. of carbon produce $2(12 + 16) = 56$ lbs. of CO, and from the table on p. 127 we see that CO weighs ·0784 lb. per cubic foot.

$\therefore$ 41·2 lbs. of carbon produce $\dfrac{56}{\cdot 0784} = 715$ cubic feet CO.

If the steam reaction follows equation ($5a$) we shall produce from 12 lbs. of carbon 48 lbs. of CO_2 and 4 lbs. of hydrogen. $\therefore$ we have $\dfrac{48}{\cdot 1225} = 382$ cubic feet CO_2 and $\dfrac{4}{\cdot 0056} = 715$ cubic feet H_2, but $41 \cdot 2 - 24 = 17 \cdot 2$ lbs. of C are used for this reaction, $\therefore$ we have $\dfrac{382 \times 17 \cdot 2}{12} = 547$ cubic feet CO_2, and $\dfrac{715 \times 17 \cdot 2}{12} = 1{,}025$ cubic feet H_2.

In reaction ($3a$) 36 lbs. of O_2 are used corresponding $\dfrac{36}{\cdot 0896} = 402$ cubic feet, corresponding to which there will be $\dfrac{402 \times 79}{21} = 1{,}510$ cubic feet N_2.

$\therefore$ Our analysis of gas becomes—

	Cubic feet.	Per cent.
CO, ...	715	18·8
CO_2, ...	547	14·4
H_2, ...	1,025	27·0
N_2, ...	1,510	39·8
	3,797	100·0

The CO will produce $56 \times 4{,}320 = 242{,}000$ B.Th.U., and the H_2 will produce $\dfrac{4 \times 17 \cdot 2 \times 52{,}500}{12} = 301{,}000$ B.Th.U., taking the "lower" calorific value for hydrogen.

∴ Altogether we have 541,000 B.Th.U. in 3,797 cubic feet of producer gas.

∴ Lower calorific value of the gas $= \frac{541,000}{3,797} = 142$ B.Th.U.

Taking the higher calorific value for hydrogen, we shall have for the hydrogen $\frac{4 \times 17 \cdot 2 \times 61,500}{12} = 352,000$, giving a total of 594,000 B.Th.U.

∴ Higher calorific value of the gas $= \frac{594,000}{3,797} = 158$ B.Th.U.

Requirements of Good Producer Plants.—The efficiency of a gas producer is measured by the ratio of the heat units in the gas produced to the heat units contained in the fuel gasified, and a high efficiency can only be obtained by giving attention to certain important factors in design and construction.

Among these may be enumerated :—

1. Adequate but not excessive grate area, giving a uniform and effective distribution of the blast to the fuel bed, the angle of incidence of the blast being adjusted to suit the diameter of the producer.
2. A deep bed of incandescent fuel of sufficient thickness to ensure a maximum reduction of the CO_2 to CO, and the complete decomposition of the steam.
3. A layer of ashes supporting the incandescent fuel and resting on the bottom of the producer lute (not on the grate bars), thus preventing the escape of unburnt carbon into the water-seal
4. Introduction of the steam saturated blast at a point where it comes into immediate contact with the incandescent fuel zone, without previously passing through the ashes which tend to obstruct its passage—a practice which in some water-sealed producers causes cooling and condensation of the steam in the blast, and consequent deterioration in the quality of the gas.
5. Accessibility to the ash zone for the purpose, with clinkering fuels, of removing clinker.
6. Facilities for evenly removing ashes from the water-sealed bottom.
7. Absolute continuity of operation, and uniformity of quality and quantity of gas produced during the cleaning of the fire.
8. Provision for the proper distribution of the fuel in the producer in order that the fuel bed may be maintained at an even thickness, thus avoiding blowholes, etc.

9. Disposal of the grate in such a manner that the greatest possible depth of fuel bed may be obtained without unduly increasing the overall height of the producer.

10. Provision for the high superheating of the steam-saturated blast and the cooling of the lining of the producer round the hot incandescent zone. This is of the greatest importance in bye-product recovery plant for high temperature furnace work where gas of the highest heating value is required, along with the simultaneous recovery of the maximum amount of sulphate of ammonia. It is also of marked advantage in dealing with fuels of a highly caking tendency, and has resulted in the successful gasification of refractory fuels which could not be worked to advantage in other types of producers.

Mond Pressure Producer.—The large type of pressure producer for use with bituminous fuel was developed largely by Dr. Ludgwig Mond.

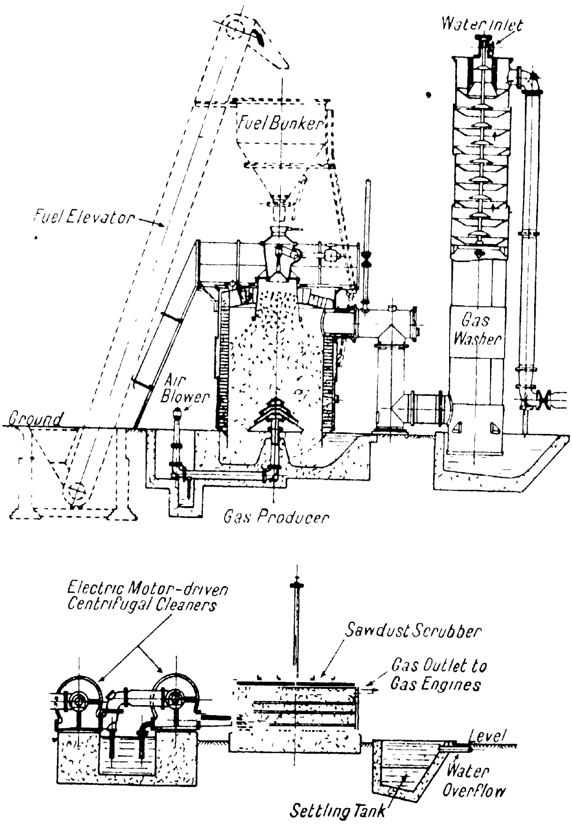
Two principal types of these pressure producers are made; one fitted with apparatus for recovering sulphate of ammonia from the coal distillate, and the one without such recovery apparatus.

The illustration shows in diagrammatic representation a modern type of non-recovery water-cooled gas producer plant as installed by the Power-Gas Corporation, Limited, for generating gas from bituminous producer coal of suitable grading, quality, and condition, for use in gas engines or for other purposes where cleanliness equal to gas engine practice is required. The arrangement of plant shown is typical of units up to about 1,100 B.H.P. capacity, larger units being generally similar, but having two or more producers, gas main connections between producers and washer to suit, and additional scrubber units.

The plant may be considered as composed of two sections, the gas-generating section—comprising the producer and its accessories—and the gas-cooling and cleaning section—comprising vertical static cooler and washer, centrifugal cleaners, and sawdust scrubber.

The producer is of the central blast type with steel casing designed to sit on concrete foundations and dip into a water-lute, formed in the foundations, and from which the ashes can be removed evenly all round without interrupting the working of the producer. The grate is conical in form, is built up of two or more C.I. sections, and provided with air slots. It is located concentrically with the producer shell, and is connected up

to the blast pipe which passes through the foundations to the outside of the producer. The lining of the producer is built



MOND PRESSURE PRODUCER.

of special quality firebricks and fireclay. The producer is fitted with a series of horizontal poke holes at about the level of the

air slots in the grate, and also with similar holes on the top, giving access to the whole of the fuel bed, these top poke holes being fitted with spherical closers.

A fuel hopper is fitted on the top of the producer, with sliding cover and inner cone valve operated by a lever and balance weight. The fuel in the hopper is discharged into a fuel container or inner bell, the top of which is provided with suitable poke holes, and from this container, automatically on to the fuel bed, this method of charging ensuring a more constant quality of gas. The gas outlet branch is fitted on the side of the producer casing close to the top.

A steel chequer-plate operating platform is arranged at the level of the top of the producer shell, with suitable protecting hand-railing and ladder giving access from ground level.

Fuel-handling plant is frequently included, and the type which is usually recommended comprises a power-driven enclosed-bucket type elevator, lifting the fuel from a boot at ground level into a bunker, which is mounted on substantial supports over the producer, and fitted with a shute furnished with lever-operated discharge valve leading down to the producer hopper. Such an arrangement of fuel-handling plant is shown in dotted lines on the illustration, but other arrangements are sometimes employed.

The air blast for the producer is provided by a blower of the steam-jet type, arranged so as to ensure a thorough intermixture of air and steam before the blast reaches the grate, but in some installations a steam-driven Roots or turbo-type blower is included, the exhaust steam from the blower engine or turbine being led into the blast pipe.

Leaving the producer, the gas enters the cooler and washer, which is of patented design. This type of washer has proved to be efficient, and the absence of moving parts, together with the freedom from possibilities of blockage, make it a reliable apparatus requiring a minimum of attention, with no upkeep charges for labour and material for packing or filling renewals such as have to be met with in most other types of washer.

From the washer the gas passes to the first of two centrifugal cleaners. These two cleaners are normally arranged to work in series, but, where desired, the requisite cross connections and valves can be added to enable either cleaner to be cut out of operation and bye-passed for purposes of overhauling should such be necessary. Each cleaner has a tar drain water-sealed

in a lute in the foundations, and the gas delivery branch of the second cleaner is joined up to a sawdust scrubber to the outlet of which the gas main leading to the gas engines, etc., is connected.

The scrubber is a rectangular cast-iron box fitted with wooden grids, on which are spread layers of wood chips, shavings, and sawdust. The gas enters at the bottom, and passing up through these layers receives its final drying and purification.

The scrubber requires very little attention, only needing to be cleaned out and refilled a few times per year, according to the load on the plant. A special lifting arrangement is provided to facilitate the removal of the cover, and all the spent material taken out can be burnt under the boiler, which provides the steam for the plant.

The drains from the various parts of the plant are run to the settling tank, which takes the form of a concrete pit with iron or reinforced concrete partition plates so arranged as to keep back the tar and allow only the water to run to the drains. The tar is then scooped or run off and disposed of in the most convenient manner, either being sold, used for boiler-firing purposes, dehydrated, or distilled, according to the quality and quantity available and local conditions.

The auxiliaries of the plant—elevator and centrifugal cleaners—can generally be most advantageously and economically driven by electric motors if electric supply is available, a direct or continuous current being preferable to an alternating-current supply.

Where requirements call for a minimum fluctuation in the pressure of the gas supply, an automatic governor can be added to plants in which a power-driven blower is installed. Such a governor would be of the gas-holder type, but of small capacity, and hence of negligible storage value. The bell of the holder is so arranged that, as it approaches its top position, it actuates a relief valve connected to the air blast main, reducing the blast supply to the producer when the gas make is in excess of the requirements, and thus automatically controlling the gas production.

In installations in which the gas generated is used to supply gas engines, and also to fire furnaces, it is advisable that a non-return valve should be provided in the gas main to the furnaces.

A suitable and sufficient supply of steam is required for blast saturation purposes; also a suitable and sufficient supply of cold water to the washer, cleaners, and producer lute. Where cold water is scarce, water-cooling and circulating plant may

be installed, and a considerable portion of the water required can be used over and over again. Suitable motive power for the auxiliaries will also be required, in the form of electricity or steam.

Some Data on Mond Gas Plants.—The calorific value of Mond gas is from 135 to 140 B.Th.U. per cubic foot; when ammonia has not been recovered from the plant a ton of good slack produces about 140,000 cubic feet of Mond gas, the effect of the ammonia recovery being to reduce this figure by about 10 per cent. The heating value of the gas represents from 75 to 85 per cent. of the total heat energy contained in the fuel used for its production.

About 60 cubic feet of gas per horse-power-hour is required with large gas engines, and the makers of the plant claim that by using Mond gas in gas engines a given quantity of fuel will produce two to three times the power obtainable from it with high-class steam engines.

"National" Suction Gas Plant.—We will now describe a typical suction gas plant made by the National Gas Engine Company, Ltd., of Ashton-under-Lyne.

The consumption of these plants per I.H.P.-hour is about 1 lb. for anthracite and $1\frac{1}{4}$ lbs. for gas coke.

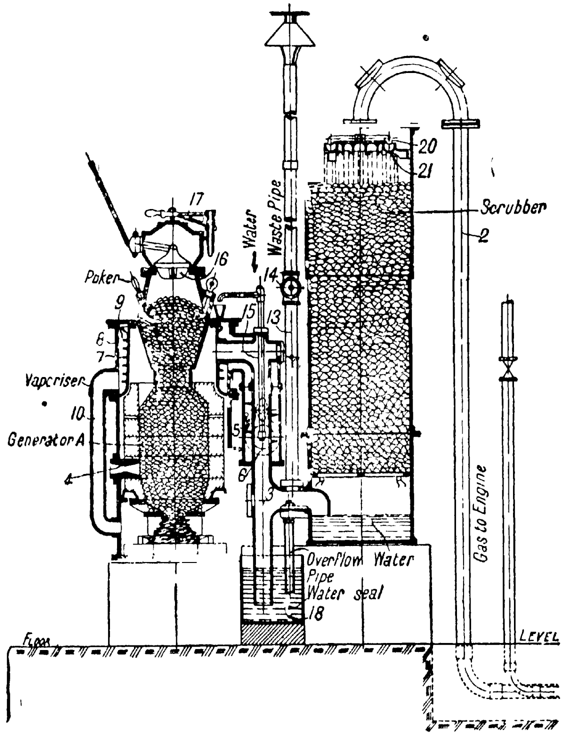
The plant consists essentially of a furnace—usually called the *Generator*—in which the fuel is burnt, and a long cylindrical vessel filled with damp coke through which the gas is passed, and consequently cleaned, on its way to the engine; this cylindrical vessel is called the *Scrubber*.

When the gas-making plant is at work in conjunction with the engine, the general action is as follows:—

(*a*) The engine draws its charge of gas from an expansion box, which is directly connected with the top of the scrubber by the gas main (2). The scrubber in turn receives its supply of gas from the gas outlet pipe (3), which connects the producer to the scrubber, and this outlet pipe is connected in such a way as to draw off from the producer the gas which is made through the partial combustion of the fuel in the furnace (4) of the producer.

(*b*) It will be thus seen that every time the engine sucks in a charge of gas, the suction action is communicated from the engine through all the interior connections of the gas plant until it is felt right at the furnace of the producer. A definite air inlet is provided to allow the air required for combustion to be drawn into the furnace of the producer at each suction stroke

so as to make additional gas to replace that drawn off by the engine, and consequently the production of gas is quite automatic, and in accordance with the demand made by the engine, which in turn is regulated by its governor.



"NATIONAL" SUCTION GAS PRODUCER PLANT.

(c) For the proper production of gas, and the good working of the producer, steam must be mixed with the air passing to the furnace, so as to keep down the temperature of the latter,

otherwise the firebars will be burnt out, and the body of the generator cracked.

(*d*) As the gas which comes off at the upper part of the producer is at a considerable temperature, it is used to vaporise the water required for the steam supply. In the National plants, therefore, the air and steam supply is arranged as follows :—

A jacket (5) is provided round the gas outlet pipe, and, under the suction effect of the engine already referred to, air passes in at the inlet (6) and, gradually circulating round the gas outlet pipe, is heated considerably before it passes into the vaporiser. The vaporiser is formed by the internal circular shell (7) and the external circular shell (8), an annular space existing between them. The inner shell (7) is heated by the outgoing hot gases coming in contact with its interior, on which heat-catching ribs are cast. On its external surface a supply of heated water is continually fed and is evaporated by the heat of the surface. There is consequently an annulus (9) which is always kept full of steam while the plant is at work.

(*e*) As soon, therefore, as the entering air, which has already been heated by its passage through the air jacket (5), reaches the vaporiser, it becomes saturated with steam in passing round the vaporiser on its way to the air and steam pipe (10). This latter accordingly feeds the space underneath the grate with a mixture of air and steam, which duly passes through the fire.

(*f*) There are a few additional important details which are required for working the plant—namely, for starting the plant —*i.e.*, before the engine is got to work; the blow-off pipe (13) is then brought into action by opening the cock (14). This pipe is extended to the outside air, and the cock (14) is shut as soon as the engine is got to work. A water heating arrangement (15) is provided, which takes further advantage of the waste heat in the outcoming gas from the producer. Double valves (16) and (17) are necessary in the coal hopper, through which the coal is introduced to the inside of the producer. Valve (16) is kept closed while the lid valve (17) is open and the fuel is poured into the coal hopper. The lid valve (17) is then replaced and the hopper valve (16) is opened, the fuel consequently dropping through. It is essential that no air should enter the producer when at work excepting in the appointed way through the air supply pipe (10), and from thence through the fire grate.

(*g*) In connection with the scrubber there is a seal pot (18), into which the overflow pipe (19) discharges the waste water from the scrubber, which is continually used whilst the plant is at work for cleaning and cooling the gas. This water is fed into the scrubber by the sprinkler pipe (20), and it is spread over the whole surface of the coke by the distributing dish (21).

LECTURE XIV.—QUESTIONS.

1. Explain by the aid of sketches the action of a Dowson **gas plant.** What is approximately the calorific value of the gas generated ?

2. Outline some form of suction gas plant. State clearly particulars of fuel and approximate composition of the gas and conditions affecting same.

3. A producer gas has the following percentage analysis by volume :— H, 16 ; CO, 20 ; CO_2, 6 ; N, 58. Determine (*a*) its calorific value per cubic foot at standard pressure and temperature ; (*b*) the minimum amount of air for complete combustion ; (*c*) the volumetric analysis of the products if combustion is complete. Calorific value of 1 lb. carbon burning to CO_2 = 14,500, burning to CO = 4,400 ; of H = 62,000 B.Th.U. Composition of air by volume, O, 21 per cent. ; N, 79 per cent. Volume occupied by 2 lbs., H = 357 cubic feet at standard temperature and pressure.

4. A gas producer, furnished with a steam jet, is working in such a manner that all the excess heat, caused by the combustion of the fuel in the presence of air to form CO gas, is utilised to cause a further reaction in which the gaseous products are CO_2 and H. Determine the composition and yield of the gas per pound of fuel, if this latter is taken as having a calorific value of 8,140 C.H.U. (14,650 B.T.U.) per lb. You may for purposes of approximate calculation assume that 1 lb. of hydrogen occupies 180 cubic feet, that the calorific value of CO is 190 C.H.U. (342 B.T.U.), and H 162 C.H.U. (291·6 B.T U.) per cubic foot. Also take air as having a composition of 20 per cent. by volume of O, and that the water is supplied at 20° C. (68° F.).

5. The volume analysis of a producer gas is :—H, 14 per cent. ; CH_4, 2 per cent. ; CO, 22 per cent. ; CO_2, 5 per cent. ; O, 2 per cent. ; N, 55 per cent. Find the air required for the perfect combustion of 1 cubic foot of the gas. If 40 per cent. excess air is supplied, find the volume analysis of the dry products. Air contains O 20·9 per cent., N 72·1 per cent. by volume.

Lecture XIV.—A M.Inst.C.E. Questions.

1. Explain with a sketch the working of a suction gas producer. What is the object of admitting vapour with the air? Describe any method of regulating the vapour supply.

2. Describe with a sketch one of the following:—(*a*) A suction gas producer; (*b*) a pressure gas producer. Also state the advantages of the type you describe.

3. Describe by sketches and explain as fully as you can the principal features of one of the following:—(*a*) The construction and method of operation of any modern gas producer, or (*b*) some type of large power gas engine.

4. Give sketches showing the construction of a suction gas-producer plant, and explain its action. What advantages does a suction producer possess over the pressure type of producer?

LECTURE XV.

THE DE LAVAL STEAM TURBINE.

CONTENTS.—Steam Turbines—Definition of a Turbine—Hydraulic and Steam Turbines—Reaction Turbine—Hero's Steam Engine—Impulse or Kinetic Energy Turbines—De Laval Steam Turbine—Conical Nozzles—Velocities of Outflowing Steam—Diagrammatic Explanation of the Sudden Changes in Pressure and Velocity in the De Laval Nozzles—Arrangement of Nozzles—Most Efficient Speed of Buckets—Example I.—Steam Consumption per Horse-Power for a Perfect De Laval Turbine—Stresses in the Material of a Turbine Wheel—Section of Small Wheels—Method of Balancing the Rotating Parts—Resistance due to Surrounding Medium—Example II.—Details of Turbine Wheel and Gearing for the De Laval Steam Turbine—Speed Reducing Gear—Lubrication of Bearings—Number of Steam Nozzles—Governor—Speed Variations—Results of Tests showing the Percentage Savings in Steam when using Superheated Steam with Varying Loads—Various Applications—Questions.

Steam Turbines.—The recent conspicuous successes of the De Laval and the Parsons Steam Turbines have revived public interest in the direct production of rotary motion, by means of the expansive properties of high-pressure steam, and taught engineers, that this—the first known method of its application—can compete successfully both in regard to economy and certainty of action with the best forms and designs of reciprocating steam engines. Prior to 1885, steam turbines were

generally looked upon as merely interesting "steam eaters," and chiefly useful for occupying the time, the faculties, and the money of sanguine imaginative inventors! The successful introduction of dynamo machines, together with the rapid development of electrical engineering during the last decade of the previous century, with its ever-increasing demands for good, high-speed, direct-coupled steam engines, have so stimulated the believers in "*the ideal rotary engine*," that, now the tide of scientific and practical opinion has turned, young engineers would do well to devote time and thought to studying the principles and action of the latest and best forms of steam turbines.

Definition of a Turbine.—*A turbine is a machine in which a gradual change of the momentum of a fluid is applied to produce the rotation of the motor.* Water and steam are the fluids most commonly used for this purpose.

Steam turbines in common with reciprocating steam engines are heat engines, converting the calorific energy of the steam into mechanical energy. From another point of view they are analogous to hydraulic turbines, and form part of the general class called "turbo-machines." With a knowledge of the thermodynamic properties of steam, and of the hydraulic turbine, it should be easy to follow with confidence, the design, construction and action of steam turbines, provided the necessary working coefficients for the calculations have been previously determined.

In turbines, the expansion of the steam can be carried to extreme limits, much more conveniently than in reciprocating engines, and hence the great advantage of employing condensers which will give the best possible vacuum. On the other hand, the fluid friction and defective shunted leakage of the steam increase in proportion as the pressure is raised. From these two opposite conditions it will be seen, that for low steam pressures, turbines are more advantageous than reciprocating steam engines, while they generally consume more steam than the latter when the back pressure of the exhaust is equal to or higher than that of the atmosphere.

Hydraulic and Steam Turbines.—Water turbines form a special type of enclosed hydraulic motor, which occupies less space, is more efficient, more easily governed, rotates at a greater speed, and is applicable to greater ranges of "head" or pressure than an ordinary water wheel. They are classified in several different ways, depending upon the manner in which their

respective special construction and action are considered. For example, we may divide them into (1) inward flow; (2) outward flow; (3) parallel or axial flow; and (4) mixed flow turbines.*

Steam turbines may be conveniently divided into four types —(1) reaction; (2) impulse or action; (3) fall of potential or continuous expansion; and (4) the "multicellular" turbines, or a combination of (2) and (3). We shall now illustrate and describe one of each of these kinds of steam motors.

(1) **Reaction Turbine—Hero's Steam Engine.**—The earliest known use of steam for producing motion was in 130 B.C., when Hero of Alexandria, in Egypt, applied the flame from a fire as at F, to heat water in a cauldron C, for the purpose of generating steam and conducting the same by a pipe P, to a globe G, from which it then issued by two oppositely directed nozzles N_1, N_2, fixed at right angles to the axis of the globe upon which it freely rotated. The reactions due to the two unbalanced pressures of the steam as it issued from the two nozzles, formed a "couple" which was capable of turning the globe at a very high speed, but with little power for a great expenditure of steam. Consequently, this simple reaction type of steam turbine has not been successfully applied as a prime mover, although many persons have tried in various ways to adapt it to the driving of small light machines.

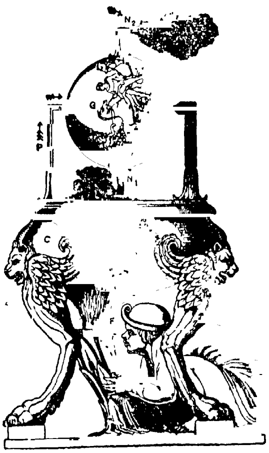


FIG. 1.—HERO'S ENGINE, 130 B.C.

(2) **Impulse or Kinetic Energy Turbines.**—In this type of steam turbine, the heat energy of the steam is first converted into kinetic energy, and then so applied to the movable parts of the turbine as to produce and keep up continuous rotation. The second part of this action is similar to that performed by water

* See the Author's *Text-Book on Applied Mechanics and Mechanical Engineering*, Vol. IV. on *Hydraulics*, Lecture V., for these kinds of turbines.

in turbines of the Pelton wheel class, although the first part does not appear in hydraulic motors. Water as used in turbines is practically of constant density, volume, and temperature, whereas the density, volume, and temperature of steam may be varied within very wide limits Water, in such cases, may therefore be regarded as having a constant weight per cubic foot, and hence the kinetic energy given up by each cubic foot, depends solely upon its effective pressure.

Now, although the weight of a cubic foot of steam is always small, still the freely-issuing steam has such a very high velocity, even at moderate pressures, that it is imbued with much greater kinetic energy than a cubic foot of water acting under the same pressure per square inch. For example, as shown by the calculation in the footnote, we see that when 1 cubic foot of steam at 35 lbs. pressure by gauge, issues freely from a pipe, it acquires *the same kinetic energy as* 1 cubic foot of water, also issuing freely from a pipe, under a pressure of 62·5 lbs. by gauge; although, the latter weighs 520 times as much as the former. Yet, because the steam turbine can have a working velocity of 2,200 feet per second, whilst the water motor can only have that of 96 feet per second, due to its "head," their available kinetic energies are practically equal.*

De Laval Steam Turbine.—The best and most successful example of the kinetic energy or impulse turbine which we can select for illustration, is that first patented in this country in 1889, by Dr. De Laval. About seven years previously, this inventor had brought out in Sweden a purely reaction type of turbine, somewhat after the style of the moving part of Hero's engine, for the direct driving of cream separators. In this, his earliest form, the rotating part simply consisted of a horizontal **S** pipe, which was freely hung by a spindle, fixed to the upper

* Since 1 cubic foot of water weighs 62·5 lbs., a vertical column of water of 1 square inch cross-section, and 144 feet in height, would occupy 1 cubic foot and cause a pressure of 62·5 lbs. on its 1 square inch base. Now, since $v^2 = 2gh = 2 \times 32 \times 144 = 96^2$, $\therefore$ the velocity of free issue $v = 96$ feet per second. Again, referring to the Steam Table in Lecture VII. of this book, we see that 1 cubic foot of steam at 50 lbs absolute or 35 lbs. pressure by gauge, weighs 0·12 lb., and from experiments or calculations of the energy in this volume of steam, it is found that the working velocity of outflow may be 2,200 feet per second. Hence, we can make the following comparison of the kinetic energy of 1 cubic foot of water and 1 cubic foot of steam at the aforesaid pressures :—

$$\left.\begin{array}{r}\text{Kinetic energy of 1 cubic foot of}\\ \text{water at 62·5 lbs. per square inch}\end{array}\right\} = \frac{Wv^2}{2g} = \frac{62{\cdot}5 \times 96^2}{2 \times 32} = 9{,}000 \text{ ft.-lbs.}$$

$$\left.\begin{array}{r}\text{Kinetic energy of 1 cubic foot of}\\ \text{steam at 35 lbs. per square inch}\end{array}\right\} = \frac{Wv^2}{2g} = \frac{0{\cdot}12 \times 2{,}200^2}{2 \times 32} = 9{,}000 \text{ ft.-lbs (approximately)}$$

and middle balanced centre of this pipe. Steam from a boiler passed into a hole on the lower side of the **S** pipe, immediately opposite the centre of motion, and passing along the bends in each direction it issued from the open ends, thus causing the bent pipe with its spindle and the cream separator connected to the latter, to revolve at a high speed, in the same manner as the old "Scotch Barker's Mill" water turbine. The low efficiency and small power obtainable from this primitive, simple arrangement, however, led De Laval to devise the present ingenious and interesting form of impulse steam turbine, which is now made in this country by Greenwood & Batley, Limited, of Leeds, to whom I am indebted for the following illustrations.

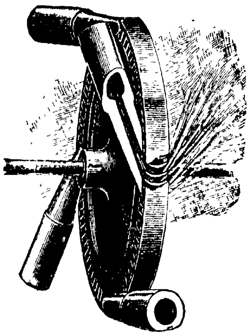


FIG. 2.—SHOWING THE ACTION OF STEAM IN DE LAVAL'S TURBINE WHEEL.

The accompanying figure shows four diverging nozzles with one cut open, to graphically exhibit the action of steam blowing or impinging upon the concave vanes, formed around the periphery of the turbine wheel. The shape of the interior of the outer ends of these nozzles is such, that the steam expands from their throats to their points in such a way as to obtain as complete adiabatic expansion as possible, and thus directly convert static heat energy into kinetic energy. This kinetic energy of the steam jets is therefore directly imparted to the little buckets of the turbine wheel, and thus causes the same to rotate at the very great working velocities shown in the following table:—

SPEEDS OF DE LAVAL TURBINE WHEELS.

Sizes of Turbine.	Middle Diameter of Wheel.	Revolutions per Minute.	Peripheral Speed.
H.P.	Mm. Ins.		Feet per second.
5	100 or 4	30,000	515
15	150 ,, 6	24,000	617
30	225 ,, 8⅞	20,000	774
50	300 ,, 11¾	16,400	846
100	500 ,, 19¾	13,000	1,115
300	760 ,, 30	10,600	1,378

The steam, after performing its work on the wheel, may be allowed to exhaust into the air. But, when a convenient supply of condensing water is available, the power, efficiency, and economical working of the turbine is greatly enhanced by connecting it with an ejector condenser. This form of condenser has been found equally applicable to small as well as to large motors, which is not the case with the more expensive and complicated air pump and condenser.

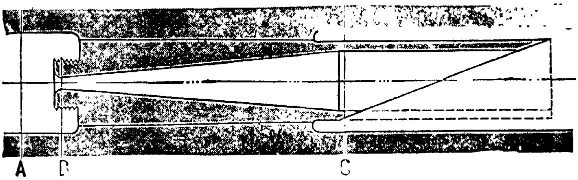


FIG. 3.—SECTION OF THE DE LAVAL STEAM NOZZLE.

Conical Nozzles.—As has just been mentioned, a very high velocity of the steam is obtained by passing it through specially shaped conical nozzles. In these nozzles the steam is expanded from its original pressure down to that in the casing in which the turbine revolves. The accompanying Fig. 3 shows a longitudinal section of a nozzle drawn to scale as first bored and reamed out to the shape shown by the dotted lines at the nose. This nose is then cut to an angle of 20°, as shown by the full front inclined line on Fig. 3, and also by Figs. 2, 4, 5, and 8. The nozzle is intended for an initial steam pressure by gauge of 200 lbs. per square inch and a vacuum of 28 inches; which means, that the steam is expanded in the nozzle from 215 lbs. absolute down to ·93 lb. absolute. During this expansion the steam (which leaves the nozzle as a cylindrical jet) attains a very high velocity. Extensive calculations have been made and experiments have proved, that if the steam is expanded adiabatically inside the nozzle, the whole of its potential energy is converted into kinetic energy, and the energy of this steam is absolutely the same as if it had been expanded in the cylinder of an engine.

Velocities of Outflowing Steam.—The kinetic energy of any moving mass is expressed by the well-known formulæ—

$$E_K = \frac{W v^2}{2g}.$$

Where E_K = Energy of kinetic form in ft.-lbs.
„ W = Weight of the mass in lbs.
„ v = Velocity of the mass in feet per second.
And, g = The acceleration of gravity in ft. per sec. per sec.

Now, suppose W to be 1 lb. of dry saturated steam of an absolute pressure p_1 lbs. per square inch, and that it is expanded adiabatically in a De Laval nozzle down to a pressure p_2 lbs. per square inch, the following equation is obtained :—

$$\left\{\begin{array}{l}\text{Heat energy at pres-}\\\text{sure } p_1 \text{ lbs. per sq.}\\\text{inch}\end{array}\right\} = \left\{\begin{array}{l}\text{Heat energy at}\\\text{pressure } p_2 \text{ lbs.}\\\text{per sq. inch}\end{array}\right\} + \left\{\begin{array}{l}\text{Kinetic energy, } E_K\text{, at}\\\text{pressure } p_2\text{, expressed}\\\text{in heat units}\end{array}\right\}$$

Or, which is the same thing :—

$$\left\{\begin{array}{l}\text{The kinetic energy, } E_K\text{,}\\\text{at pressure } p_2\text{, ex-}\\\text{pressed in heat units}\end{array}\right\} = \left\{\begin{array}{l}\text{Heat energy at}\\\text{pressure } p_1 \text{ lbs.}\\\text{per sq. inch}\end{array}\right\} - \left\{\begin{array}{l}\text{Heat energy at pres-}\\\text{sure } p_2 \text{ lbs. per sq.}\\\text{inch}\end{array}\right\}$$

If H_1 and H_2 be the heat energy at pressures p_1 and p_2 lbs. per square inch respectively, when expanding adiabatically, and J the mechanical equivalent of heat, then, since $E_K = W\,v^2/2\,g$, and since $W = 1$ lb., we get, $E_K = 1 \times v^2/2\,g = v^2/2\,g$. Or, $v = \sqrt{2\,g\,E_K}$.

But, by Lec. XIII., Vol I., the kinetic energy E_K or total work to be got out of each lb. of steam, must equal the heat-energy in the same; or, $E_K = J\,(H_1 - H_2)$ ft.-lbs. Hence, substituting this value of E_K in the above equation, $v = \sqrt{2\,g\,E_K}$; we get—

$$\left.\begin{array}{r}\textit{The velocity of the}\\\textit{outflowing steam,}\end{array}\right\}\; v = \sqrt{2\,g\,J\,(H_1 - H_2)}. \qquad \text{(I.)}$$

Consequently, *every pound* of dry saturated steam which issues from a boiler through a smooth, rounded circular hole and then expands adiabatically, is imbued with—

$$\text{The } \textit{Kinetic Energy}, \; E_K = \frac{v^2}{2\,g} = J\{(t_1^\circ - t_2^\circ) + L_1 - x\,L_2\}.^* \quad \text{(II.)}$$

Where, x = The specific quantity or dryness of steam in 1 lb. of the fluid.
„ t_1° = *Temperature* of the initial steam at pressure p_1 absolute, from Table II., Vol. I.

*If the student refers back to Table II., Lecture VII., and to Lectures IX., XI., and XIII., Vol. I., he will see how—

$$H_1 = S_1 + L_1 \quad \text{and} \quad H_2 = S_2 + xL_2.$$
$$\text{Or,} \quad H_1 = t_1 + L_1 \quad ,, \quad H_2 = t_2 + xL_2.$$
$$\therefore \; H_1 - H_2 = t_1 + L_1 - (t_2 + xL_2) = (t_1 - t_2) + L_1 - xL_2.$$

Where, t_2° = *Temperature* of the exhaust steam at pressure p_2 absolute, from Table II.

„ L_1 = *Latent Heat* of steam at pressure p_1 lbs. absolute, from Table II.

„ L_2 = *Latent Heat* of steam at pressure p_2 lbs. absolute, from Table II.

As stated in a previous lecture, the internal heat depends upon the dryness fraction of the steam, consequently the calculations are made on the basis of constant entropy during its adiabatic expansion. It has been proved by experiments, that it is only when a certain ratio exists between the steam pressures p_1 and p_2, that the maximum amount of steam flows through a converging nozzle; and, further, that this maximum flow always takes place if these circumstances exist. This ratio for saturated steam is—

$$\frac{p_2}{p_1} = 0{\cdot}577\ ; \quad \text{or, } p_2 = 0{\cdot}577 \times p_1.$$

If, for instance, we have dry steam at an initial pressure $p_1 = 215$ lbs. absolute, at the inlet A, as shown by Figs. 3 and 4.

And, if the percentage of moisture by weight in the steam = 0.

„ if the specific quantity of the steam = 1 lb.

„ if the specific volume of the steam = 2·11 cub. ft. per lb.

Then, at the line B, which is drawn at the throat or smallest section of the nozzle, we will obtain from the above equation—

$$\text{The pressure } p_2 = {\cdot}577 \times p_1 = {\cdot}577 \times 215$$
$$= 125 \text{ lbs. pressure absolute.}$$

Now, due to the change in the steam from mere potential energy at A to that of partial kinetic energy at B, we find—

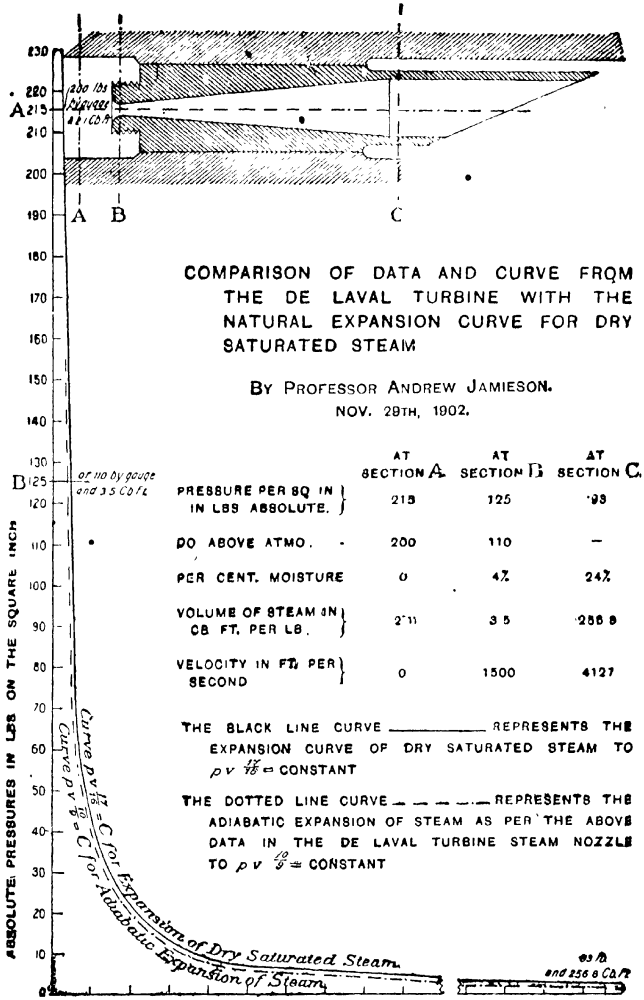
The percentage of moisture by weight in the steam	=	4 per cent.
The specific quantity of steam . . .	=	·96 lb.
The specific volume of steam . . .	=	3·5 cub. ft. per lb.
The velocity of steam	=	1,500 ft. per second.

Further, if at the largest section, C, where the steam leaves the nozzle, we get—

The pressure	=	·93 lb.
The percentage of moisture by weight in steam	=	24 per cent.
The specific quantity of steam . .	=	·76.
The specific volume of steam . .	=	256·8 cub. ft. per lb.
Velocity of steam	=	4,127 ft. per second.

It will thus be seen, that the object of the diverging part of the nozzle is to further expand the steam. For this initial steam pressure and vacuum the proportion between the areas of the smallest and largest sections of the nozzles should be as 1 to 27·23. If the nozzle be properly constructed, the steam leaving it has the same pressure as the surrounding medium, and as no further expansion takes place it must leave the nozzle in a cylindrical jet. This jet impinges on the vanes of the turbine-wheel placed before the nozzle, and, as the radial length of the buckets is always a little larger than the diameter of the jet, all the steam leaving the nozzle must pass into and out of the buckets.

Diagrammatic Explanation of the Sudden Changes in Pressure and Velocity in the De Laval Nozzles.—Since the sudden fall of pressure and temperature of the steam, when travelling such a short distance into the nozzle, appears to be more or less of a paradox to young engineers, the following diagram and explanation have been reproduced from the *Proceedings of the Institution of Engineers and Shipbuilders in Scotland*, as given therein by the author at the discussion on the De Laval turbine. This explanation was put forward with the view of trying to explain, not only the sudden fall of pressure near to and in the throat of the steam nozzle of the De Laval turbine, but also the further fall of pressure along the conical part to its mouth. The full line curve represents the natural loss of pressure in dry saturated steam as it expands in accordance with Professor Rankine's well-known formula, $p\,v^{\frac{17}{16}}$ = a constant; where p is the pressure in lbs. per square inch absolute, and v the corresponding volume in cubic feet per lb. of steam. This curve may be drawn by students to a large scale, from 475 lbs. per square inch—at which pressure, 1 lb. of the steam occupies 1 cubic foot—down to 1 lb. absolute, at which it occupies 330 cubic feet. But, in the reduced figure, it only includes the range of pressures specially mentioned in this example. The dotted line represents an adiabatic expansion curve to $p\,v^{\frac{10}{9}}$ = a constant, from 215 lbs.



COMPARISON OF DATA AND CURVE FROM THE DE LAVAL TURBINE WITH THE NATURAL EXPANSION CURVE FOR DRY SATURATED STEAM

BY PROFESSOR ANDREW JAMIESON.
NOV. 29TH, 1902.

	AT SECTION A	AT SECTION B	AT SECTION C
PRESSURE PER SQ IN IN LBS ABSOLUTE.	215	125	·93
DO ABOVE ATMO.	200	110	—
PER CENT. MOISTURE	0	4%	24%
VOLUME OF STEAM IN CB FT. PER LB.	2·11	3 5	256 8
VELOCITY IN FT. PER SECOND	0	1500	4127

THE BLACK LINE CURVE ———— REPRESENTS THE EXPANSION CURVE OF DRY SATURATED STEAM TO $pv^{\frac{17}{16}}$ = CONSTANT

THE DOTTED LINE CURVE — — — —·— REPRESENTS THE ADIABATIC EXPANSION OF STEAM AS PER THE ABOVE DATA IN THE DE LAVAL TURBINE STEAM NOZZLE TO $pv^{\frac{10}{9}}$ = CONSTANT

absolute at A, before entering the nozzle. down to 0·93 lb. at C, where it occupied 256·8 cubic feet, and leaves the nozzle with a velocity of 4,127 feet per second with 24 per cent. of moisture. This curve passes through the point, B, where the steam occupies 3·5 cubic feet, and has 4 per cent of moisture, with a velocity of 1,500 feet per second. Now, it is evident from this curve, that if the *potential energy* of each lb. of static steam at A has been *so far converted* into *kinetic energy* at B, that its velocity is 1,500 feet per second, and contains 4 per cent. of moisture, with an increase of volume from 2·11 cubic feet at A to 3·5 cubic feet at B, it *must of necessity* have fallen in pressure from 215 to 125 lbs. absolute pressure in doing so! This is in strict accordance with the natural law for the adiabatic expansion of steam. The temperature of the steam must also fall from 382° Fah. to at least 340° Fah. in this short passage, and *must thereby* loose *potential energy* due to friction and increased velocity. This

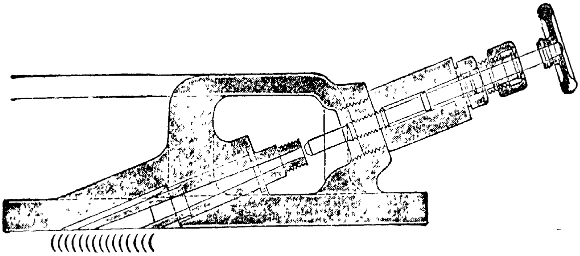


FIG. 5.—ARRANGEMENT OF NOZZLE AND SHUTTING-OFF VALVE.

accounts for its expansion from 1 lb. of dry saturated steam at A, to that of slightly moist steam at B, with a corresponding and natural loss of pressure and temperature due to the partial transformation of its *potential energy* at A, into *kinetic energy* at B, with the certified velocity of 1,500 feet per second. All the references to steam injectors and to ejectors, &c., as found in some books and papers, do not help the student to arrive at a reasonable solution of this problem. The simple fact remains, that steam has been proved to undergo such changes, and when these are coupled with a reference to the adiabatic curve of expansion, they seem quite sufficient to account for the sudden drop in pressure up to B. Also, the further increase of volume and velocity down to its entry into the turbine-wheel buckets, with almost all its potential energy and expansive properties taken out of it—when viewed by aid of the adiabatic curve—is a clear and complete solution of this, to some, an apparent paradox.

Arrangement of Nozzles.—Having arrived at the kinetic energy of the steam, the next point is to arrange that the largest possible amount of this kinetic energy shall be utilised by the vanes of the wheel for the production of mechanical energy. In the De Laval turbine, the nozzle is placed at an angle of 20° to the plane of rotation. This is shown by the accompanying figure, and there is a space of only $\frac{1}{16}$ inch between the face of the nozzle and the buckets of the wheel. Consequently, there is no loss of velocity in the steam jet between its leaving the nozzle and its entering the buckets of the turbine wheel.

Most Efficient Speed of Buckets.—The most efficient speed for the buckets is clearly that speed at which the steam will leave them at the lowest relative velocity. The maximum efficiency is obtained when the peripheral speed of the turbine wheel (*i.e.*, the linear velocity of the buckets), runs at 47 per cent. of the velocity of the outflowing steam, when the angle between the nozzle and the plane of rotation of the wheel is 20°. The absolute velocity of the steam leaving the buckets is then 34 per cent. of its initial velocity, and since the kinetic energy given up by the steam is proportional to the square of the velocities, we get—

$$\text{The efficiency, } \eta = \frac{v_1^2 - v_r^2}{v_1^2} = \frac{100^2 - 34^2}{100^2} = 88 \text{ per cent.}$$

Where, v_1 = Velocity of the steam impinging on the blades or vanes of the turbine = (say) 100.

And, v_r = Velocity of the fluid when leaving the ring of vanes = 34, as compared to v_1 = 100.

This means, that when a wheel is running at 47 per cent. of the velocity of the outflowing steam, 88 per cent. of the total kinetic energy of the steam is absorbed by the turbine wheel for the production of mechanical energy.

The best speed for any De Laval wheel can be easily calculated from the above rule. In the nozzle of the previous example, where the steam pressure was 215 lbs. per square inch absolute, the vacuum was 28 inches or ·93 lb. absolute, and the velocity of the steam jet 4,127 feet per second, from which we get the correct speed of the centre line of the buckets to be (Fig. 4 and p. 244)—

$$V_1 \times 47 \text{ per cent.} = \frac{4{,}127 \times 47}{100} = 1{,}940 \text{ feet per second.}$$

This tremendous speed is about $22\frac{1}{2}$ miles per minute, and is far too high for any practical purpose at present. Moreover, it

VELOCITY OF OUTFLOW, KINETIC ENERGY AND H.P. OBTAINABLE FROM DRY SATURATED STEAM WITH THE DE LAVAL TURBINE.

Initial Steam Pressure. Lbs per Square Inch by Gauge	Counter Pressure, 1 Atmosphere.			Counter Pressure, 2·4 Lbs. per Square Inch Absolute, Corresponding to 25-Inch Vacuum.			Counter Pressure, 0·93 Lb per Square Inch Absolute, Corresponding to 28-Inch Vacuum.		
	Velocity of Outflow of Steam Feet per Second.	Kinetic Energy Foot-Lbs per Second	H.P of 550 Foot-Lbs. per Second.	Velocity of Outflow of Steam Feet per Second	Kinetic Energy. Foot-Lbs. per Second	H.P. of 550 Foot-Lbs. per Second	Velocity of Outflow of Steam Feet per Second.	Kinetic Energy. Foot-Lbs per Second	H.P. of 550 Foot-Lbs per Second.
		Per Lb of Steam per Hour.			Per Lb of Steam per Hour.			Per Lb of Steam per Hour	
60	2,421	25·29	0·046	3,320	47·57	0·087	3,680	55·44	0·106
80	2,595	29·06	0·053	3,423	50·56	0·092	3,793	62·08	0·113
100	2,717	31·86	0·058	3,520	53·47	0·097	3,871	64·66	0·118
120	2,822	34·37	0·062	3,596	55·80	0·101	3,940	66·99	0·122
140	2,913	36·62	0·066	3,661	57·84	0·105	3,999	69·01	0·125
160	2,992	38·63	0·070	3,718	59·65	0·108	4,045	70·61	0·128
180	3,058	40·35	0·073	3,764	61·14	0·111	4,091	72·22	0·131
200	3,115	41·87	0·076	3,810	62·64	0·114	4,127	73·50	0·134
220	3,166	43·26	0·079	3,852	64·03	0·116	4,159	74·64	0·136
280	3,294	46·83	0·085	3,962	67·74	0·123	4,229	77·18	0·140

On looking at the Kinetic Energy columns in this table, it will be noticed what an enormous increase in power is obtained from the same quantity of dry steam when exhausting into a condenser with a good vacuum, as compared with the result obtained when exhausting the steam directly into the atmosphere.

can only be attained in a machine like the De Laval turbine, where the wheel revolves quite freely in its casing, and where there are no rubbing surfaces or steam-tight joints to contend against. The highest speed yet made use of is about 1,380 feet per second, in the case of the 300 H.P. De Laval turbine (p. 236)

When any fluid, such as wet, dry or superheated steam, air or any other gas expands adiabatically from a vessel at pressure p_1 lbs. per square inch (wherein it has *no* velocity), into a place where its pressure is p_2 lbs. per square inch, then we can find the work which it would do if admitted to an ordinary cylinder with piston, but with no clearance, when expanding adiabatically to the same final pressure p_2. We know that for each 1 lb. of the fluid the work done by it is equivalent to its change from potential to kinetic energy, or $\frac{v^2}{2g}$. It is also equal to the number of B.T.U. given up by the steam during its adiabatic expansion multiplied by Joule's equivalent.

Hence, from the previous equations (I.) and (II.) in this lecture and the above, we get—

$$E_K = \frac{W v^2}{2g} = J(H_1 - H_2) = J\{(t_1^\circ - t_2^\circ) + L_1 - x L_2\}.$$

Where $W = 1$ lb. and $v = \sqrt{2gJ(H_1 - H_2)}$, we get—

$$\sqrt{2gJ} = \sqrt{2 \times 32{\cdot}2 \times 778} = 224.$$

$$\therefore \quad v = 224\sqrt{(H_1 - H_2)} \text{ feet per second.}$$

$$\text{Or,} \quad v = 224\sqrt{(t_1^\circ - t_2^\circ) + L_1 - x L_2} \text{ feet per second}$$

EXAMPLE I.— Find the velocity with which dry saturated steam of 215 lbs. absolute issues from a boiler into a De Laval nozzle, wherein it expands adiabatically to a terminal pressure of 1 lb. per square inch. Here let the dryness fraction = ·76 at the point of discharge from the nozzle upon the turbine buckets —*i.e.*, let the steam have 24 per cent. of moisture, or all exactly as in the diagram, Fig. 4.

Referring to the above equation and to Table II., Lecture VII., Vol. I., for the various known values, we get—

$$t_1^\circ = 387{\cdot}7, \quad t_2^\circ = 102, \quad L_1 = 838{\cdot}9, \text{ and } x L_2 = {\cdot}76 \times 1{,}043.$$

$$\text{Hence } v = 224\sqrt{(t_1^\circ - t_2^\circ) + L_1 - x L_2}.$$

$$\text{Or} \quad v = 224\sqrt{(387{\cdot}7 - 102) + 838{\cdot}9 - ({\cdot}76 \times 1{,}043)},$$

$$\therefore \quad v = 224\sqrt{332} = 224 \times 18{\cdot}2 = 4{,}076 \text{ feet per second.}$$

Whereas, $v = 4{,}127$ in the facing table, issued by the makers of the De Laval turbine, or only 1·3 per cent. more than our calculation.

Steam Consumption per Horse-Power for a Perfect De Laval Turbine.—If, for instance, the speed of the steam entering the buckets of the turbine wheel be 4,000 feet per second, the speed of the steam leaving the buckets should be 1,360 feet per second, and the horse-power per lb. of steam—

$$\frac{v_1^2 - v_f^2}{2g \text{ foot-lbs. per second}} = \frac{4{,}000^2 - 1{,}360^2}{2g \times 550 \times 3{,}600} = \cdot 11 \text{ H.P.}$$

And the steam consumption per H.P.-hour would be—

$$\frac{2g \times 550 \times 3{,}600}{4{,}000^2 - 1{,}360^2} = 9{\cdot}1 \text{ lbs.}$$

Stresses in the Material of a Turbine Wheel.—The stresses in the rotating materials (more especially those due to centri-

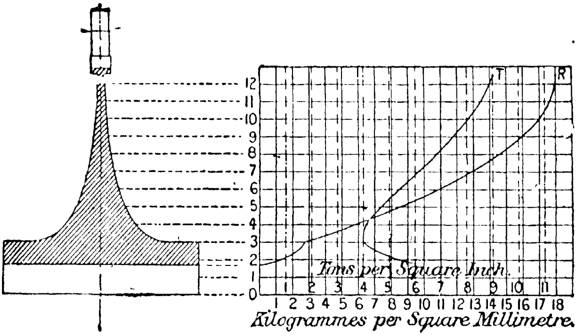
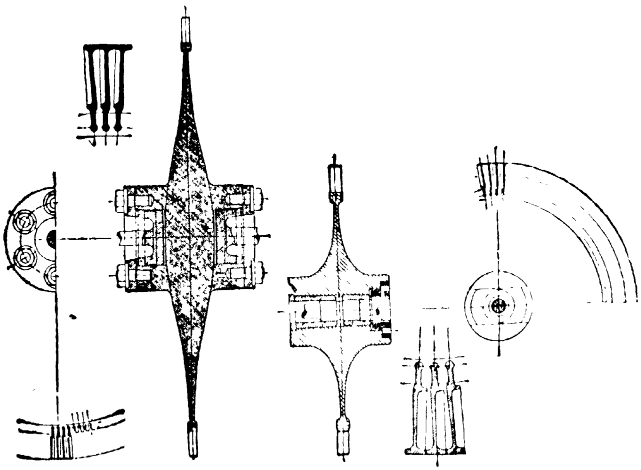


FIG. 6.—STRESSES IN THE MATERIAL OF A DE LAVAL TURBINE WHEEL.

fugal force), are much greater than are usually met with in practical engineering. To give some idea of the magnitude of these stresses, it may be mentioned that in the 300 H.P. turbine the centrifugal force caused by each bucket weighing 250 grains is 15 cwts., or 47,000 times the statical stress produced by the mere weight of the bucket, when the wheel is revolving at its normal velocity! The tangential stresses also increase towards the hole through the centre of the wheel. To avoid this, the larger sizes are made solid without a hole through their central boss, but the shaft is made in two pieces and fixed to the wheel by flanges and screws, as shown by Fig. 7. To withstand the centrifugal stresses the wheel is made of a solid disc with the buckets dovetailed around its circumference.

Section of Small Wheels.—The stresses in the wheel are tangential and radial. Consequently, if we call the radial stress R, and the tangential stress T, it will be evident that both R and T, increase with the radius, and are greatest at the circumference of the wheels. Further, these stresses depend on the axial thickness of the wheel in each place, and they also affect one another. It will be seen from the diagram (Fig 6), which shows the stresses in a wheel for a 50 H P. steam turbine, that the wheel is so constructed that both stresses R and T have



LARGE WHEEL. SMALL WHEEL.

FIG. 7.—SECTIONS SHOWING ARRANGEMENT OF TURBINE WHEELS.

their largest value at the circumference of the wheel, just where the buckets are fixed Hence, the wheel is not made of uniform strength, but is strongest at the centre which is the heaviest part. The various cross sections of the wheel are therefore so proportioned to meet the case Consequently, for the weakest part of the wheel, where the buckets are fixed, a recess is turned in the outer portion of the wheel, so that if it should burst from an excess of speed, it would give way at this recess, and the vanes would become detached, when the wheel would stop rotating. The vanes are so light that no damage would ensue.

In designing a turbine wheel, a certain coefficient of safety is used, and fixed stresses in the material at different diameters of the wheel are adopted. These stresses are so proportioned that breaking would take place if the wheel was run at about double the number of revolutions required in actual work.

Method of Balancing the Rotating Parts.—The great difficulty here is, that no matter how carefully the disc is balanced when in a state of rest, its centre of gravity would not identically coincide with the geometrical centre round which the wheel revolves. At a very high speed this would cause such severe vibrations, that no ordinary bearings would be able to withstand the stresses arising therefrom. This difficulty was overcome by using a flexible shaft. On account of the very high speed of the wheel, a shaft of very small diameter is sufficient to transmit the power. The bearings supporting this shaft are naturally fixed at a considerable distance on each side of the wheel to give the shaft freedom. The vibrations, however, increase with the speed up to a certain point, which is called "the critical speed of the wheel," at which speed they suddenly disappear, and the wheel settles down and runs perfectly smoothly in its bearings. This phenomenon is known as the "settling of the wheel," and it is caused by the wheel rotating at the slower speeds round the geometrical centre, but when it reaches the "critical speed" the shaft bends a little out of the geometric centre line; thus, the wheel automatically begins to rotate round its true centre of gravity.* On account of the flexibility of the shaft, and the extreme accuracy of the turning and balancing of the wheel, this settling takes place at from $\frac{1}{5}$ to $\frac{1}{8}$ of the maximum normal running speed of the wheel. In fact, in the modern turbines, this effect is hardly noticeable, as all the revolving parts of the turbine are most carefully balanced, by the parts being mounted on the shafts with tapered centres.

* The reason for the above phenomenon cannot be scientifically explained, but assuming, as is very probable, that the settling of the wheel occurs when the number of revolutions per minute is equal to the number of vibrations which the shaft makes with the wheel mounted upon it, then, the critical speed can be calculated, and it is found to be—

$$N = k\sqrt{\frac{F}{W}}.$$

Where N = number of revolutions per minute.
F = the force required to bend the shaft a certain distance.
W = the weight of the turbine wheel.
k = constant.

This formula seems to correspond very closely with the results obtained by actual tests.

Resistance Due to Surrounding Medium.—From the very beginning of experiments with this turbine, Dr. De Laval found it necessary to adopt very high speeds if the machine was to be constructed on the "action" principle. A high linear velocity of the buckets of the power-wheel of the turbine, is only to be obtained either by using a small wheel running at a great number of revolutions, or by employing a larger wheel running at a comparatively slower speed. There are, however, two points to be taken into consideration in the question of wheels or bodies revolving at a high rate of speed. One is the strength of the material of which the wheel is to be constructed, and the other is the resistance of the surrounding medium to the motion of the wheel due to surface friction. Another matter of equal importance is the bulk and weight of the machine. It is found that the resistance of the turbine wheel increases more rapidly with the diameter of the wheel than with the number of revolutions, and for this reason, and on account of the bulk and weight, small wheels running at high speeds are used for machines of small power, and larger wheels running at a modified speed for larger turbines. As the question of economy is becoming of more importance, the size of the wheels and also the number of revolutions in the larger unit of machine are so proportioned, that with the increasing size of the units the velocity of the vanes of the wheel approach more closely to what it ought to be from a theoretical point of view. The resistance to which the revolving wheel is subjected from the surrounding medium depends partly on the skin friction, and partly on churning or eddy making. It is found in practice that this resistance is almost exactly proportional to the density of the surrounding medium, and that it increases approximately with the fifth power of the diameter and the third power of the number of revolutions. It will be evident that the thinner the medium which surrounds the wheel the less will be the resistance offered to its motion. The resistance is less in dry saturated steam than in air of the same pressure, and it decreases as the vacuum becomes more and more perfect; also, the resistance is less in superheated than in saturated steam, and it decreases with the amount of superheating.

EXAMPLE II.—A 150 H.P. turbine wheel is subjected to a resistance of 35 H.P. when running in steam of one atmosphere (or 30 inches of mercury absolute pressure), but if run in a vacuum of 28 inches (2 inches of mercury absolute pressure), the resistance will be decreased in about the same proportion.

Or, $30'' : 2'' :: 35 \text{ H.P.} : x \text{ H.P.} = 2\frac{1}{3} \text{ H.P.}$

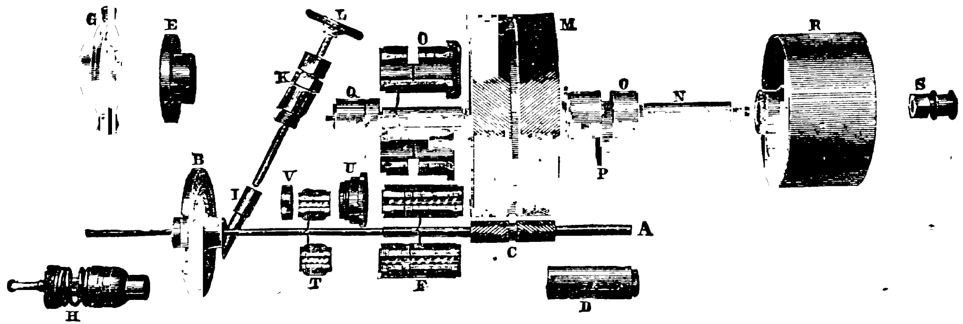


FIG. 8.—DETAILS OF TURBINE WHEEL, FLEXIBLE SHAFT, BEARINGS, AND GEARING FOR THE DE LAVAL STEAM TURBINE.

INDEX TO PARTS OF STEAM TURBINES.

A for Turbine shaft.
B ,, Turbine wheel.
C ,, Pinion.
D ,, End bush.
E ,, Safety bearing, in turbine box.
F ,, Middle bush, in two parts.
G ,, Safety bearing, in box cover.
H ,, Ball bush, with adjusting spring.

I for Steam nozzle
K ,, Stuffing-box, with stop valve to nozzle.
L ,, Wheel to valve spindle.
M ,, Gear-wheel.
N ,, Gear-wheel shaft.
O ,, Gear-wheel shaft bushes, in two parts.

P for Lubricating ring.
Q ,, Centrifugal governor.
R ,, Driving pulley.
S ,, Stopnut with pulley, for connecting to tachometer.
T ,, Tightening bush, in two parts.
U ,, Adjusting nut, with spring.
V ,, Friction gland.

This is a gain of $32\frac{2}{3}$ H.P. The velocity of the steam jet issuing into a vacuum is, moreover, higher than the velocity of outflow into the atmosphere, and both the above circumstances make it essential that turbine machinery should be run under as good a vacuum as can be maintained.

Speed-Reducing Gear.—The best speeds of these wheels for economy and efficiency of steam are far too great for direct coupling to ordinary machines, and, therefore, it is necessary to reduce the same to speeds generally in use. This is done by means of double helical gearing. The pinion is made of very hard steel, while the teeth on the large wheel are of softer steel. The speed of the gearing—that is, the linear velocity of the teeth—is about 100 feet per second.

In the small size turbines there is only one gear wheel (Figs. 8, 9), but in the larger sizes there are two—one wheel being on each side of the pinion—with, of course, two low speed shafts, which prevent any great side pressure on the high speed shaft, as shown by Fig. 10. The gear is generally arranged for a reduction of 10 to 1, and works very well. In the larger sizes it is accompanied, when working, by a dinging or hissing noise, to which there is little objection.

What may be the life of this gear is not yet known, but it must be great when lubricated with good oil, since after four or five years regular working it is impossible to see any signs of wear. In one or two of these turbines the only signs of wear after a few months work were, that the driving side of the teeth had a better polish than the other side.

Lubrication of Bearings.—The high speed bearings are usually lubricated with sight feed lubricators, while the low speed shaft bearings are lubricated with the now universal ring system of oiling, as shown by Fig. 9.

Number of Steam Nozzles.—Small turbines are fitted with only one steam nozzle, but the number is increased in proportion to the size of the turbine. Each nozzle has its own stop valve. Consequently, if the turbine has to deal with a light load, one or more of the nozzles can be turned off, so that the remaining nozzles may work at their best efficiency (see Figs. 5, 8, and 10).

The Governor.—The governing of these turbines is effected by means of a very simple type of centrifugal governor, and is attached to the end of the low speed shaft. The two balls or expanding parts are supported on knife edges, and work with very little friction. The centrifugal action of the governor balls

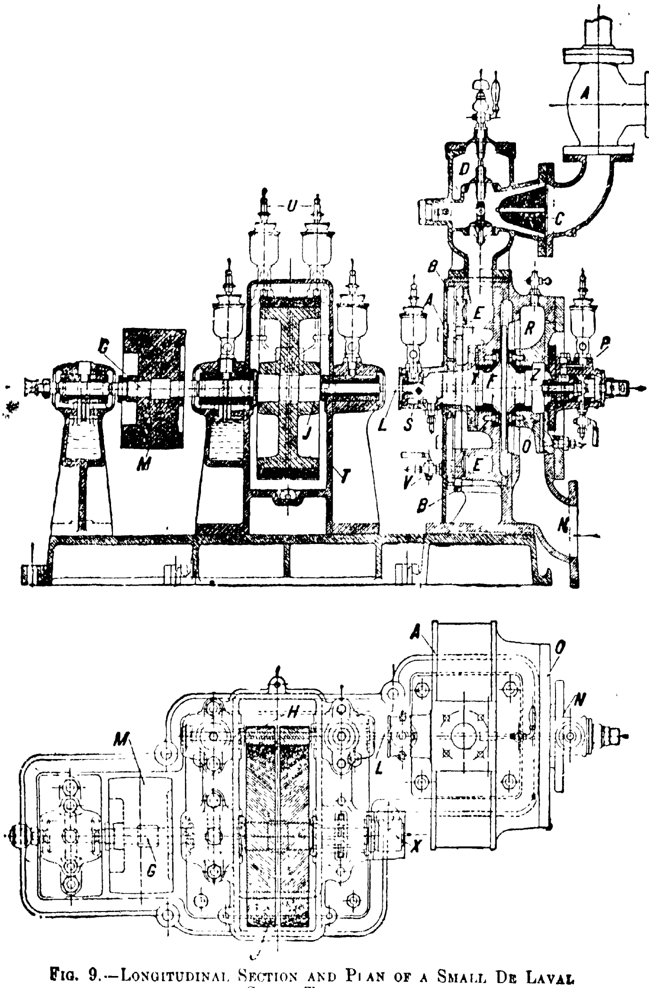


FIG. 9.—LONGITUDINAL SECTION AND PLAN OF A SMALL DE LAVAL STEAM TURBINE.

forces out a central pin, which presses on a lever, and thus actuates the governor valve.

Speed Variations.—De Laval turbines work with any steam pressure between 50 and 200 lbs. square inch, when exhausting either into the atmosphere or into a condenser. The only change required for doing so is in the nozzles, which are interchangeable, and are shaped differently for the different pressures, and according to the amount of expansion of the steam. Sometimes turbines are fitted with two sets of nozzles,—one set for discharging into the atmosphere, and one set for exhausting into a condenser vacuum.

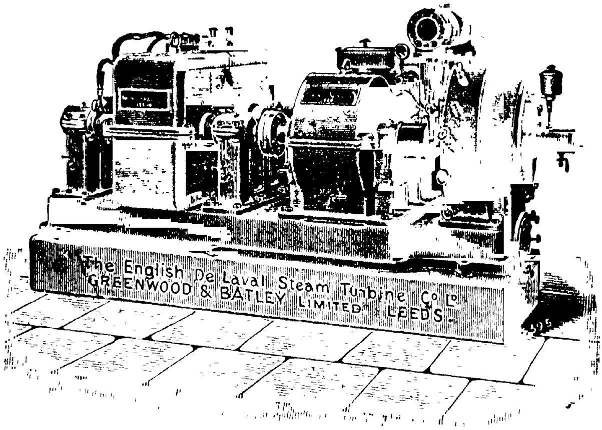


FIG 10.—ONE HUNDRED HORSE-POWER DE LAVAL DYNAMO PLANT.

THE DE LAVAL PATENT STEAM TURBINE.
Index to Parts in Fig 9.

A for Steam stop-valve.
B ,, Cover for steam chest.
C ,, Steam sieve.
D ,, Governor valve.
E ,, Steam chest.
F ,, Turbine wheel.
G ,, Gear-wheel and pulley shaft.
H ,, Pinion.
J ,, Gear-wheel.
L ,, Flexible shaft.
M ,, Belt-pulley.
N for Exhaust.
O ,, Cover for turbine-case.
P ,, Lubricator for ball-bush.
R ,, Exhaust passage.
S ,, Tightening-bush on flexible shaft
T ,, Gear-case.
U ,, Lubricators for gearing.
V ,, Drain-cock for steam chest.
X ,, Centrifugal governor.
Y Z ,, Safety bearings.

Results of Tests.—The velocity of the steam when leaving the nozzles increases with the initial pressure, but as we have already seen the greatest gain in velocity is obtained by exhausting into a high vacuum. For example, referring to table on p. 530 in this lecture, we see, that with an initial pressure of 100 lbs. by gauge, and exhausting into the atmosphere, the velocity is 2,717 feet per second; whereas, when exhausting into a 28-inch vacuum it reaches 3,871 feet per second. This means, that with a 28-inch vacuum there is a saving of nearly 50 per cent. in the steam used, because in each case the kinetic energy varies directly as the square of the velocity of impact of the steam on the turbine wheel. Superheating the steam is advantageous to the turbine, as it gives the steam jet a higher velocity, thus increasing the kinetic energy of the steam, and it also diminishes the resistance to the rotation of the turbine wheel. Any degree of superheating can be used with De Laval turbines, as the highly superheated steam does not come into direct contact with the moving parts of the machinery. By the time the steam reaches the chamber in which the turbine wheel revolves, it has already attained the pressure and temperature of the exhaust steam. It has been found from the results of tests that not only the steam consumption, but also the heat consumption in B.T.U. per H.P.-hour are lowered when the turbine is driven with highly superheated steam, as shown by the following table:—

TABLE SHOWING THE PERCENTAGE SAVINGS IN THE DE LAVAL TURBINE WHEN USING SUPERHEATED STEAM FOR LOADS REQUIRING THE FULL USE OF EIGHT, SEVEN, AND FIVE NOZZLES RESPECTIVELY.

No of Nozzles open.	Steam Pressure in Lbs by Gauge below Governor	Vacuum in Inches	Amount of Superheat at Governor Valve.	B.H P. Load with Superheated Steam.	B H.P. Load with Saturated Steam.	Lbs. of Superheated Steam used per B.H P.	Lbs. of Saturated Steam used per B.H.P.	Percentage gain by use of Superheated Steam.
								Per Cent.
8	198·5	27 2	84° F.	352	333	13·94	15·17	8·8
7	197	27·4	64° F	298·4	284·8	14·35	15·56	8·4
5	197·7	27·4	16° F.	196	195·2	15·53	16 54	6·5

The illustration (Fig. 10) shows the application of a 100 H.P. De Laval turbine to the driving of a twin armature

high speed dynamo. The speed is regulated by a sensitive governor of the centrifugal type, and the steam consumption has been proved to be only 24·5 lbs. per electrical horse-power-hour (or say about 20 lbs. per I.H.P.-hour) in the case of a 50 horse-power Turbo-dynamo plant, supplied with steam at 114 lbs. per square inch and with an ejector condenser making a vacuum of 13 lbs. per square inch.

Various Applications.—This steam turbine can be used for a variety of purposes :—

1. It may be arranged for driving mill machinery by belting or ropes, in which case the turbine should be bolted to the foundation, but in many other cases this seems to be unnecessary.

2. It is extensively used for driving dynamos, as shown by Fig. 10, and in the larger sizes there are always two armatures. This also has the advantage, that the machine may be used for two different voltages by coupling the armatures either in series or in parallel, which is suitable for a three-wire system of electrical distribution.

3. These turbines are frequently used for the direct driving of centrifugal fans and pumps. Their high speed renders this combination very efficient. In the case of having to pump water up from mines or to higher elevations, two or more pumps may be coupled in series. The turbine fan can deliver air of high pressure, and the machine can, therefore, be used as a blower for cupola furnaces with great advantage.

Lecture XV.—Questions

1. Express in your own words the definition of a turbine Give a list of the different types into which hydraulic and steam turbines may be conveniently classified. State clearly how, from one point of view, the steam turbine fulfils the same functions as the reciprocating steam engine, whilst from another point of view it acts like a water turbine.

2 Sketch and describe any form of reaction turbine. Give reasons why this same principle has not been successfully applied to steam turbines.

3. Explain clearly the difference between an impulsive hydraulic motor and an impulsive steam turbine Give a comparative pair of numerical examples similar to that in the lecture, but with water of 100 lbs pressure per square inch and steam of the same kinetic energy.

4. Show by neat sketches the passage of steam along the conical nozzle and buckets of the De Laval turbine. Why and to what angle is the interior of the steam nozzle made conical? Why and to what angle is the nose of the nozzle cut after it has been bored and reamed?

5. Give in your own words a clear, diagrammatic explanation of the sudden changes in pressure and velocity of the steam in passing through the nozzles of the De Laval turbine. Explain in concise detail how the final formula for the kinetic energy of the steam leaving the nozzle is arrived at, and show how the velocity of the outflowing steam is calculated.

6 Suppose that 1 lb. of dry saturated steam of 160 lbs absolute pressure per square inch be expanded adiabatically in a De Laval nozzle down to an absolute pressure of 1 lb. per square inch, and let the dryness fraction = ·76 at the point of discharge from the nozzle upon the turbine buckets. Find the velocity with which the steam leaves the mouth or Section C of the conical nozzle. Draw a corresponding diagram to Fig. 4, with a table to illustrate your answer, and let $\frac{1}{16}$ inch represent 1 lb. pressure per square inch, as well as 1 cubic foot of volume per lb. of steam.

7. Sketch and describe the usual arrangement of nozzle with shutting-off valve. State any advantages or disadvantages which you consider may be brought forward in favour of or against this method of regulating the steam supply to the turbine wheel.

8. Illustrate by a numerical example, how you would arrive at the most efficient form of the buckets. Enumerate any drawbacks to the direct driving of machines by the De Laval turbine, and explain clearly how the speed of the wheel is reduced by gearing for driving dynamos, fans, or centrifugal pumps.

9. Plot out to a large scale upon squared paper, the stresses in the material of a De Laval turbine wheel. Why are the larger sizes of wheels made solid without a hole through their central boss? Give sections showing how the shaft is fixed to the boss in both large and small wheels.

10. Describe the method of balancing the rotating parts of a turbine. Also, explain the phenomena known as the "critical speed" and the "settling of the wheel." Give a formula by which the "critical speed" or number of revolutions per minute of the wheel may be calculated.

11. Give detailed sketches of the several parts, with a complete index, for the De Laval steam turbine.

12. Give a longitudinal section and plan of a small De Laval steam turbine, with the necessary letters and index to parts. Show how the moving parts are lubricated, and state what alterations are required when you wish to exhaust into a condenser instead of into the atmosphere.

13. Plot out on squared paper the tabulated results of tests taken from a De Laval turbine (1st) when using superheated steam, (2nd) when using saturated steam as given in the table, p. 254

14. Enumerate the various applications for which the De Laval turbines are specially adapted.

LECTURE XVI.

PARSONS, CURTIS, AND OTHER STEAM TURBINES.

Mathematical Explanation of How the Heat Units, Work Done, and Change of Momentum are expressed for Ideal Steam Engines, with Special Reference to Steam Turbines.*—Referring to the formula used in connection with the De Laval turbine, it will be interesting and instructive to examine more minutely, the ideal exchange of heat units for work done by steam, and apply the same to continuous expansion turbines.

Heat Units which should be given out per Lb of Steam in an Ideal Engine, when Exhausting into the Atmosphere.*—Neglecting the variation of the specific heat of water, the usual formula for the total heat in steam, or total heat of evaporation as given in Lecture IX., Vol. I., is as follows:—

$$H = S + L.$$

Or, $$H = (t^\circ - 32^\circ) + 966 - \cdot 7\,(t^\circ - 212^\circ),$$

i.e., $$H = 1{,}082{\cdot}4 + \cdot 3\,t^\circ \text{ (from } 32^\circ \text{ F.).}$$

Where t° = the temperature in Fahrenheit degrees.

For arithmetical convenience, the total heat of evaporation calculated from zero Fahrenheit will be got by adding 32 to 1,082·4 as above.

* See footnote on next page.

Hence, $\quad \mathrm{H} = 1{,}115 + \cdot 3\, t^\circ$ (*from* 0° *Fah.*).

Now, taking the absolute temperature as $\tau = (461 + t^\circ \text{ Fah.})$, and adapting the above equation to this absolute temperature, we get—

$$\mathrm{H} = 1{,}082{\cdot}4 + \cdot 3\, t^\circ.$$

Or, $\quad \mathrm{L} = (\mathrm{H} - \mathrm{S}) = (1{,}082{\cdot}4 + \cdot 3\, t^\circ) - (t^\circ - 32^\circ).$

„ $\quad \mathrm{L} = 1{,}082{\cdot}4 + 32 - t^\circ + \cdot 3\, t^\circ.$

„ $\quad \mathrm{L} = 1{,}115 - \cdot 7\, t^\circ.$

„ $\quad \mathrm{L} = 1{,}115 - \cdot 7\,(\tau - 461).$

„ $\quad \mathrm{L} = 1{,}115 - \cdot 7\,\tau + 322{\cdot}7.$

Hence, The Latent Heat in Steam, $\quad \mathrm{L} = 1{,}438 - \cdot 7\,\tau$ (*from abs. zero of temp.*).

Or, The Total Heat in Steam, $\quad \mathrm{H} = 1{,}438 + \cdot 3\,\tau$ (*from abs. zero of temp.*).

Heat Units which should be given out per Lb. of Steam in an Ideal Engine by Expanding the Steam Adiabatically and then Exhausting into a Condenser.*—A formula will now be deduced from the first principles of thermodynamics, by which the heat units due in the shape of work done per 1 lb. of steam, may be easily calculated therefrom. It is understood, that this steam is of the same quality for which Regnault determined the latent heat, as shown above.

Let 1 lb. of water be heated from temperature τ_1 to temperature τ_2, and converted into steam at that temperature τ_2. The steam is now *expanded adiabatically* in the *ideal engine* until its temperature has fallen to the starting temperature τ_1. Then, let it be wholly condensed while the *temperature remains constant* at τ_1. Heat is absorbed by the water at *all* temperatures between τ_1 and τ_2, and also the heat equivalent to the latent heat of steam is being absorbed at temperature τ_2, during the whole process of the formation of steam. Heat is neither taken in nor given out during the adiabatic expansion of the steam, but is given off at temperature τ_1, during condensation. If a minute quantity of heat $d\,\mathrm{Q}$, be taken in by the water at *any* temperature τ (somewhere between τ_1 and τ_2), then the work obtainable from the steam due to that quantity of heat is—

$$\frac{d\,\mathrm{Q}\,(\tau - \tau_1)}{\tau}.$$

* Engineering students should study Paper No 2,306, "Economy Trials of a Non-Condensing Steam Engine: Simple, Compound and Triple," by Peter William Willans, M.Inst C E, in the *Proceedings of the Institution of Civil Engineers*, vols. xciii., xcvi, and cxiv., where Mr. Willans, Dr. John Hopkinson, and others discuss the following and its effects.

The heat absorbed in raising 1 lb. of water from τ to $(\tau + d\,\tau)$ was $d\,Q$, and the work to be obtained from it is, therefore—

$$\frac{d\,Q\,(\tau - \tau_1)}{\tau}.$$

Hence, the work obtained from the heat, applied to raise the temperature of the water from τ_1 to τ_2 is—

$$\int_{\tau_1}^{\tau_2} \frac{d\,Q\,(\tau - \tau_1)}{\tau} = \int_{\tau_1}^{\tau_2} d\,Q - \tau_1 \int_{\tau_1}^{\tau_2} \frac{d\,Q}{\tau} = (\tau_2 - \tau_1) - \tau_1 \log_e \frac{\tau_2}{\tau_1}.$$

The heat absorbed in evaporating 1 lb. of water at temperature τ_2, is already shown to be $(1{,}438 - \cdot 7\,\tau_2)$, and the work obtainable therefrom, by working to a lower temperature τ_1, is therefore found by multiplying this quantity by the efficiency limit of the Carnot cycle, as shown in Lecture XIII.

Or, Work done as expressed in heat units for the Latent Heat of steam (*only*) $= (1{,}438 - \cdot 7\,\tau_2)\left(\frac{\tau_2 - \tau_1}{\tau_2}\right).$

Therefore, taking τ_2 as the absolute temperature of the steam in the steam-chest, and τ_1 as the absolute condenser temperature in degrees Fahrenheit, the total heat units due from 1 lb. of saturated steam expanding adiabatically is clearly the sum of the work done due to the *Latent* and *Sensible Heats*.

$$\text{Total Heat} = \overbrace{\text{B.T.U.} = (1{,}438 - \cdot 7\,\tau_2)\left(\frac{\tau_2 - \tau_1}{\tau_2}\right)}^{\text{Latent Heat}} + \overbrace{\left\{(\tau_2 - \tau_1) - \tau_1 \log_e \frac{\tau_2}{\tau_1}\right\}}^{\text{Sensible Heat.}}.$$

It is worthy of notice, that this is not the maximum efficiency working between temperatures τ_2 and τ_1. To attain this, the steam must not be wholly condensed at temperature τ_1, but it must be so far condensed, that on compressing the steam adiabatically, it should be raised to precisely the temperature τ_2, when it was wholly condensed, but this is impracticable in the steam engine.

A slight modification of the last formula based on the assumption, that when 1 lb. of water at temperature τ_1 is raised in temperature to τ_2 as water, it receives all its heat $(\tau_2 - \tau_1)$ at the average temperature, $\frac{\tau_2 - \tau_1}{2}$, may be written thus—

$$\text{B.T.U.} = \left\{\left(\frac{1{,}438 - \cdot 7\,\tau_2}{\tau_2} + \frac{\tau_2 - \tau_1}{\tau_2 + \tau_1}\right)(\tau_2 - \tau_1)\right\}.$$

This equation is known as MacFarlane Gray's formula, which he deduced from Rankine's cycle and a study of entropy dia-

grams. It has been applied to the Willans & Robinson central valve engine, as well as to the Parsons steam turbine. The Hon. Mr. C. A. Parsons says, "that he gets from 60 to 65 per cent. of the possible work done, as calculated by this formula, in his steam turbines." It is applicable to the previously-described De Laval turbine, and gives the velocities of the outflowing steam closer to those in the table of Lecture XV., than by the formula and the example in that lecture.

Example I.—Find the velocity of the steam issuing from the nozzle of the De Laval turbine, when the pressure of the entering steam is 215 lbs. per square inch absolute, expanding adiabatically and leaving the nozzle at a pressure of ·93 lb. per square inch absolute by the previous formula.

Looking up Table II. on the Properties of Saturated Steam, and Fig. 4 in the previous lecture, we get the temperature of 387·7° F. for steam at 215 lbs., and 100° F. for steam at ·93 lb. absolute pressure per square inch. Using MacFarlane Gray's formula, we obtain the total heat units, which is equivalent to the work done, as follows:—

$$\text{Total B.T.U.} = \left\{\left(\frac{1{,}438 - \cdot 7\,\tau_2}{\tau_2} + \frac{\tau_2 - \tau_1}{\tau_2 + \tau_1}\right)(\tau_2 - \tau_1)\right\}$$

$$\text{,,} \quad = \left\{\left(\frac{1{,}438 - \cdot 7 \times 848\cdot 7}{848\cdot 7} + \frac{848\cdot 7 - 561}{848\cdot 7 + 561}\right)(848\cdot 7 - 561)\right\}$$

$$\text{,,} \quad = \left\{\left(\frac{1{,}438 - 594}{848\cdot 7} + \frac{287\cdot 7}{1{,}409\cdot 7}\right)(287\cdot 7)\right\}$$

$$\text{,,} \quad = \left\{\left(\frac{844}{848\cdot 7} + \frac{287\cdot 7}{1{,}409\cdot 7}\right)(287\cdot 7)\right\}$$

$$\text{,,} \quad = \{(\cdot 99 + \cdot 2)(287\cdot 7)\}$$

$$\text{,,} \quad = 1\cdot 19 \times 287\cdot 7$$

$$\text{,,} \quad = 342 \text{ per lb. of steam between } \tau_1 \text{ and } \tau_2.$$

Referring to the previous part on the De Laval turbine, we get the formula for the velocity in feet per second of the steam as—

$$v = 224\sqrt{H_1 - H_2},$$

$$v = 224\sqrt{342},$$

$$v = 224 \times 18\cdot 49 = 4{,}142 \text{ feet per second.}$$

It will be noticed, that this result (4,142) agrees more closely with that given in the table just after Fig. 5 of Lecture XV. (viz., 4,127) than what was arrived at by using the other formula (viz , 4,076), but both are sufficiently near for practical purposes.

Continuous Expansion Steam Turbines.—In this type, the gradual change of momentum of the fluid takes place, due to a loss of pressure or potential in the steam as it flows from the inlet to the outlet of the turbine. The passage of the steam may either be parallel with the axis of the cylinder, in which case it is called a *parallel-* or *axial-flow* turbine, as shown in Fig. 1; or, it may pass radially towards and from the axis alternately, when it is termed a "*radial-flow*" turbine. In either case, the same weight of steam enters and leaves the motor per second, but the volume naturally increases whilst the temperature and pressure fall. The route followed by the steam through, and the mechanical action of, this motor are, however, not very difficult to understand, even in the latest compound condensing engines.

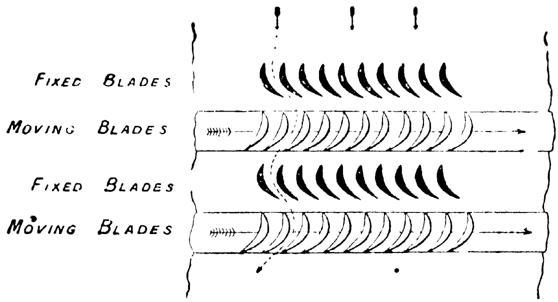


FIG. 1.—ACTION OF STEAM ON THE BLADES OF A PARSONS' TURBINE.

Parsons' Steam Turbine.—Ever since 1884 the Hon. C. A. Parsons, F.R.S., has been continuously engaged in perfecting and applying his steam turbine to the running of dynamos, fans, pumps, and fast speed vessels of various kinds. We have only room for a general description of his compound condensing parallel-flow motor, as applied to the direct driving of electric generators and screw propellers.

General Description of the Working Cylinders.—As will be seen from the accompanying Figs. 1, 2, and 3, this steam engine consists of a cylindrical case with rings of inwardly projecting guide blades, within which revolves a concentric shaft with rings of outwardly projecting blades. The blades of the cylinders nearly touch the shaft, and the blades on the shaft lie between those on the case and nearly touch the same. This annular space between the shaft and the case, is therefore fitted with

alternate rings of fixed and moving blades. Steam enters the left-hand end of this annular space at A (Fig. 3), from the boiler steam pipe S P, through the stop valve S V, into the steam chest S C_1, through the safety governor equilibrium valve S G V, shown in the left-hand partitioned steam chest. The steam now flows into steam chest S C_2, then through the double beat valve D B V, and down to the space A. From A, the steam passes to the right through a ring of fixed guide blades, by which it is projected in a rotational direction upon the succeeding ring of

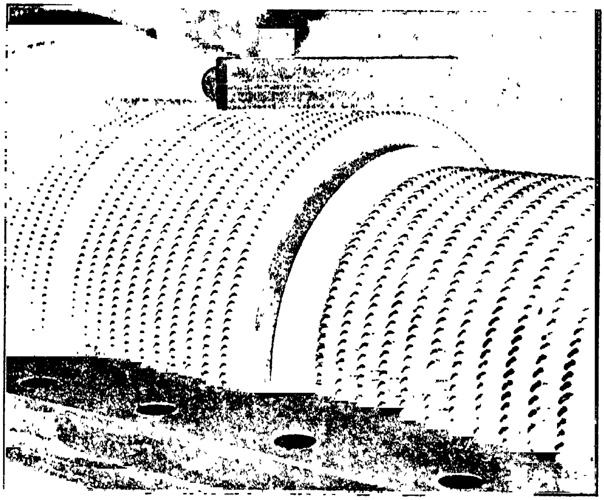


FIG. 2. PORTION OF THE INTERIOR OF THE CYLINDER OF A PARSONS' COMPOUND TURBINE, SHOWING THE HIGH, INTERMEDIATE, AND BEGINNING OF THE LOW-PRESSURE BLADES PROJECTING FROM THE SHAFT.

moving blades, imparting to them a rotational force. It then passes through the succeeding ring of guide blades, and the reaction therefrom increases the rotational force. The same process takes place at each of the successive rings of guide and moving blades. The steam therefore expands gradually by small increments through each of the three sizes of cylinders which may be termed the high, intermediate, and the low-pressure cylinders H C, I C, and L C respectively, before it exhausts through E P_2

into the air or into a condenser. In a moderate-sized turbo-motor there may be from thirty to eighty successive rings, and when the steam arrives at the last ring the expansion has been completed. On the left side of the steam inlet at A, are the three dummy or balancing pistons, H P, I P, L P, which are fixed to and rotate with the shaft Each of these pistons corresponds in size to the part of the turbine it balances, and to which it is connected by a pipe. On their outsides are grooves and rings which project into corresponding grooves in the case. The three balancing pistons H P, I P, and L P, and their projecting rings are thus kept nearly touching each other, so as to make a practically tight joint. The object of these pistons is to steam balance the three working pistons in H C, I C, and L C, and thus relieve the end pressure on the thrust bearings. The thrust bearing T B, at the free end of the shaft can be adjusted to regulate the amount of clearance between the pistons and the grooves.

Functions of the Electric and Centrifugal Governors.—When this steam turbine is coupled direct to a continuous current dynamo, or generator supplying current for electric glow lamps, where it is very important to keep the voltage as constant as possible, whatever be the load or number of lamps in circuit, then the speed of the turbine is regulated by the electric governor E G (Fig. 3). But, if the centrifugal governor C G, be also fitted, it only acts as a safety or emergency governor to close the safety governor valve S G V.

When the turbine is coupled direct to alternators, or large direct current generators, which have to be run in parallel with others for electric tramways, &c., it is very important to keep the speed constant. The turbine is then only fitted with two centrifugal governors—viz., the one for controlling purposes, whilst the other acts as a safety governor.

General Description of Steam Admission and Regulation (see Figs. 3 to 5).—Suppose the main steam stop valve S V (Fig. 3), to be fully opened, and that the engine is coupled to a continuous current dynamo, and its speed regulated by the electric governor E G; then, if the voltage rises beyond the normal, E G pulls down the right-hand end of the long lever G L, whose fulcrum is at F, and raises the left-hand end of G L. This lever lifts a small relay plunger valve R P V, which allows the steam in the relay chamber R C to escape, and the relay spring R S, above the relay piston R P, then partly closes the double beat valve D B V (Figs. 3 and 4). With this type of electric governor the speed rises as the load on the engine increases; whereas, with the centrifugal governor it falls slightly.

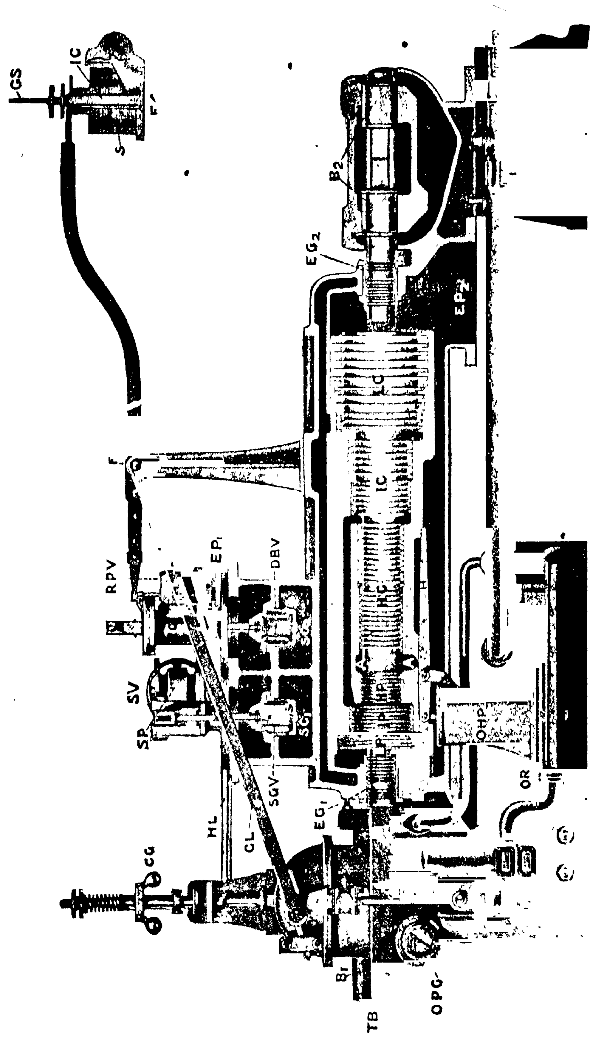


FIG. 3.—SECTIONAL ELEVATION OF PARSONS' PARALLEL-FLOW COMPOUND TURBINE, WITH GOVERNOR, EQUILIBRIUM ADMISSION AND GOVERNOR VALVES, STEAM CYLINDER AND BLADES, &C.

S V for Stop valve.
S P ,, Steam pipe.
SC₁, SC₂ ,, Steam chests 1 and 2.
S G V ,, Steam governor valve.
D B V ,, Double beat valve.
R P V ,, Relay piston valve.
R C ,, Relay cylinder.
A, A ,, Admission port.
H P, H C ,, High balance piston and cyl.
I P, I C ,, Intermediate ,, ,,

L P, L C for Low piston and cyl.
E P₁, ₂ ,, Exhaust passages.
E G₁, E G₂ ,, End glands.
B₁, B₂ ,, Bearings for cyl. shaft.
T B ,, Thrust bearing.
H, O H P ,, Handle for oil hand pump.
O R ,, Oil reservoir.
O P ,, Oil pump.
O P G ,, Oil pressure gauge.
C G ,, Centrifugal governor

H L for Hand lever for S G V.
S L ,, Starting lever for D B V (see Fig. 4).
C ,, Cam for working C L.
G L ,, Governor lever.
F ,, Fulcrum for C L and G L.
E G ,, Electric governor.
S ,, Solenoid for E G.
I C ,, Iron core for E G.
G S ,, Governor spring

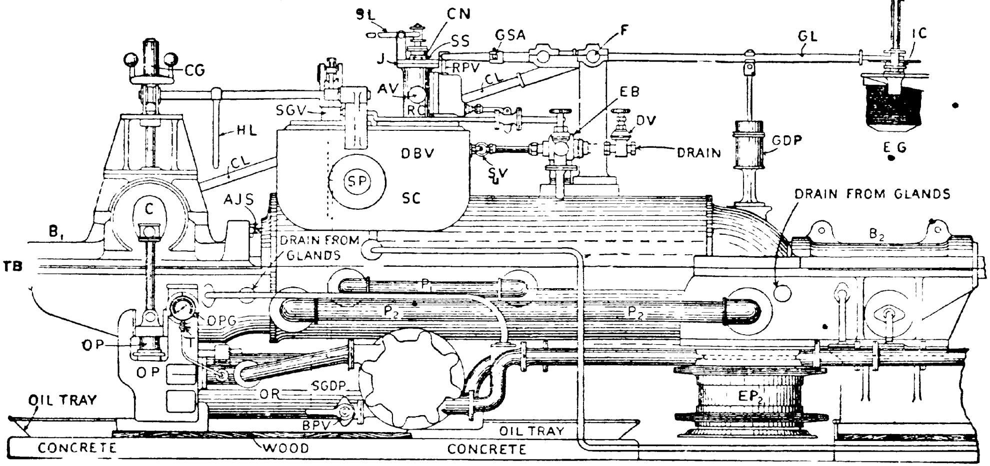


FIG. 5.—OUTSIDE LONGITUDINAL VIEW, SHOWING THE GENERAL ARRANGEMENT OF A PARSONS' STEAM TURBINE FOR DRIVING DYNAMOS.

The steam which passes from S V, through D B V, gets into the circular admission chamber at A, and acts upon the working and the balancing pistons as previously described.

Should the speed with the voltage be increased beyond the required limits, then the balls of the safety centrifugal governor C G, fly outwards and act on the horizontal rod to which the hand-lever H L, is attached. This rod, therefore, releases the trigger and allows a weight to pull down the safety double beat valve S G V, and cut off steam entirely, until the plant is started up again afresh.

Details from the Stop Valve to the Relay Piston (Figs. 3 to 5).—Steam from the boiler is admitted by the steam pipe S P, and stop valve S V, to the steam chest $S C_1$. At first, the starting lever S L, is lifted (Fig 5), when the steam passes into the steam chest $S C_2$, to the double-beat valve D B V. This valve is now lifted by the starting lever S L (Figs. 4, 5). It thereafter works automatically by the force of the steam under the relay piston R P, in the relay chamber R C. The admission valve A V (Fig. 5), regulates the amount of steam to work the relay piston R P, and is usually about half a turn open. It should not be more unless the boiler pressure is low. The motion of the relay plunger valve R P V, is derived from the action of the cam C, on the cam lever C L. On the down stroke of R P V, it closes the exhaust port $E P_1$ from chamber R C, so that when steam is turned on at the stop valve S V, the relay piston, actuated by the accumulation of steam under it, lifts the governor double-beat valve D B V, and admits steam to the engine. When the exhaust port $E P_1$ is open, the steam can escape into the atmosphere in a non-condensing turbine or into the end glands in a condensing one, and the double-beat valve D B V, is then closed by the force of the relay spring R S, over the relay piston R P.

To Warm the Engine.—See that the safety governor valve S G V, is open. Always gradually warm the engine as far as the exhaust $E P_2$, by slightly opening the main stop valve S V, and lifting the governor double-beat valve D B V, by the starting lever S L, provided for that purpose.

To Start the Engine.—Open the engine stop valve S V, a little more, and assist the relay piston R P, by the lever S L, to lift the double-beat valve D B V. Always allow the engine to run slowly until the oil is circulating freely, as shown by the small opening in the three-way cock, fitted to the oil pressure gauge O P G (Figs. 3 and 5).

Bearings and their Lubrication.—The bearings B_1 and B_2, are lubricated under pressure by the oil pump O P. The lubricant

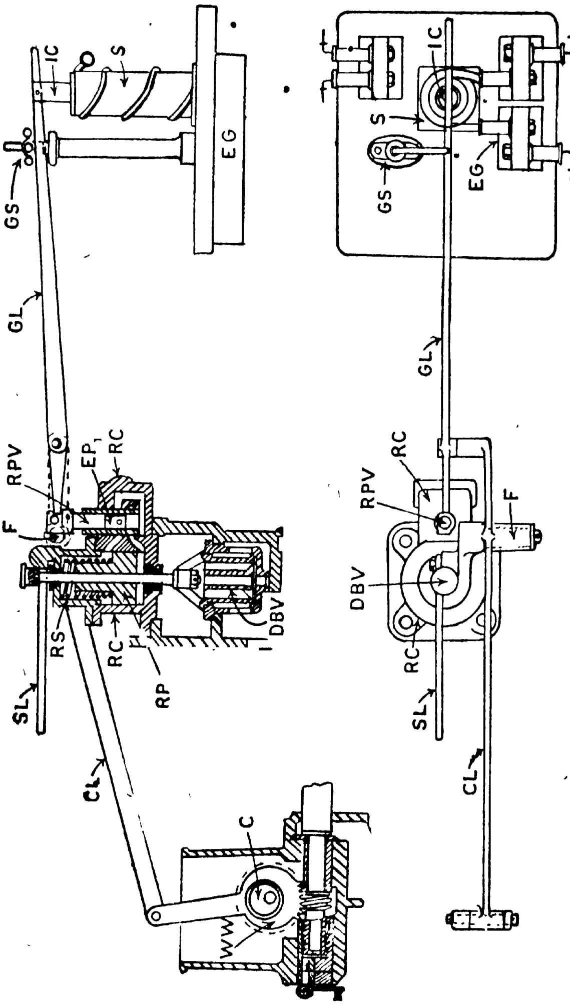


FIG. 4.—SECTIONAL ELEVATION AND PLAN OF ELECTRICAL GOVERNOR FOR PARSONS' TURBINE.

passes through the cooler in the oil reservoir O R, on its way to the bearings. Any good mineral oil of fair body, free from acid and gummy matter may be used. The practice of running off 8 to 10 per cent. of the oil each day from the drain at the bottom of the reservoir O R, and putting the same through a filter will greatly conduce to good results, prevent the accumulation of dirt, gummy deposit, water in the reservoir and oil pipes. When running, there should be 2 inches of oil shown by the oil gauge glass, and the bye-pass valve B P V, connecting the feed pipe and the oil reservoir O R should be shut. Large machines are supplied with an auxiliary oil hand pump O H P, as shown by Fig. 3, for the purpose of forcing the oil by the hand-lever H, into the bearings when starting the plant.

Electric Governor.—The relay plunger valve R P V, in Fig. 4,* is controlled by the action of the solenoid S, on its soft iron core I C, and against the force of the governor spring G S, which is supported by the holder at the end of the governor lever G L. The outlet from the relay chamber R C, is open, when the lever G L, is at the top and bottom positions of its range, and closed when in the middle position. The solenoid S, must always be permanently connected in series with a fixed and an adjustable rheostat, and the whole joined as a shunt across the main terminals of the dynamo. The dynamo field magnetising circuit must always be permanently connected to the dynamo terminals without switches or other apparatus, whereby the circuit could be broken.

The Safety or Centrifugal Governor, in Figs. 3, 5, is adjusted to act at the desired revolutions per minute. It is a good custom to shut down a plant by the hand-lever H L, which disengages the trigger, and allows the safety governor double-beat valve S G V —between the governor and screw-down valves—to close, before closing the main stop valve S V.

General Instructions for Adjusting the Governors (Figs. 3 to 5).—See that the relay spring R S (Fig. 4), has sufficient compression to close the double-beat valve D B V with full steam pressure, when the governor lever G L is in the lowest position of its range. If it does not do so, slack off the joint J, check nut C N, and increase the compression by the screwed sleeve S S. The compression is generally about $\frac{1}{4}$ inch to $\frac{5}{16}$ inch. The left end of governor lever G L, which actuates the relay plunger valve R P V, is provided with a screw adjustment G S A. Its nut should be near the centre of its range when the governor lever G L, is in the best governing position.

* Fig. 4 was obtained by the kind permission of Mr. R. M. Neilson, A.M.I.M.E., from his book on *The Steam Turbine.*

The relay plunger valve R P V, should be examined occasionally when the engine is first put to work; and, if necessary, cleaned with fine emery cloth, but should not be made to leak. This valve should be free enough to fall with its own weight, and must not bind on the rounded end of the governor lever. The governor lever dash-pot G P D, should be kept filled with paraffin oil, and the safety governor dash-pot S G D P (Fig. 5), kept filled with water.

Condensing Type.—The exhaust steam from the relay cylinder R C, is led to the end glands $E G_1$ and $E G_2$, of the turbine case to supply them with steam. There are valves on each of the two pipes P_1 and P_2, leading to the end glands, one of which must be kept full open, and the other regulated to suit the gland steam supply. This is most important, for, if one gland valve be not full open, the relay cylinder will not act properly. Always keep a trace of steam blowing at the glands, or air is liable to pass into the condenser through them.

Non-Condensing Type.—The exhaust steam from the relay cylinder R C, should be led away as free from obstruction as possible, to the atmosphere. The discharge valve D V, should be full open, and the pipe leading away from it quite free from obstruction. If steam should leak from the end glands $E G_1$ and $E G_2$, then open the supplementary valve $S_U V$ in Fig. 5, which supplies steam to the jet in the ejector E B, until the leakage ceases.

When air and circulating pumps are driven from the turbine engine shaft, the first "warming-up" of the engine should be discontinued as soon as the exhaust from $E P_2$ is warm, for otherwise the air pump will become heated, and, consequently, will act slowly.

The Brush-Parsons Turbo-Generator.—As a supplement to the foregoing illustrations and descriptions we herewith reproduce an unlettered outside plate-view of this plant with a longitudinal section of the turbine and of its shaft with the pistons, in order that the student may test his understanding of the arrangement.

It will be observed from the outside view that the cylinder with its supports are cast in one piece. These are made of hard close-grained cast iron. The shaft, as well as its rotor, are made of solid forged steel, annealed and turned all over the outer surfaces, as shown in Fig. 7. The blades are securely attached to this rotor and cylinder respectively. The blades for the high-pressure cylinder, which are naturally subjected to the effects of the superheated steam, are made of special material to withstand the action of high temperature steam.

The bearings are of gunmetal fitted with concentric tubes to allow of perfect alignment, as described under Fig. 8.

The oil pump, which supplies the continuous circulation of the lubricant under pressure for these bearings, is driven by means of a worm and wheel direct from the left-hand end of the turbine shaft, as seen in Fig. 6.

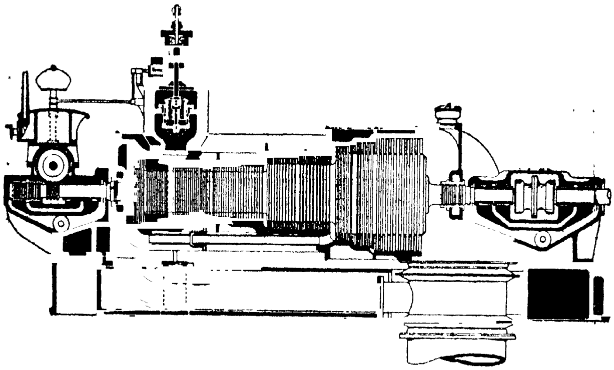


FIG. 6.—LONGITUDINAL SECTION OF A BRUSH-PARSONS STEAM TURBINE.

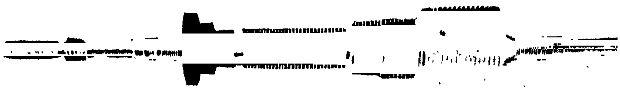
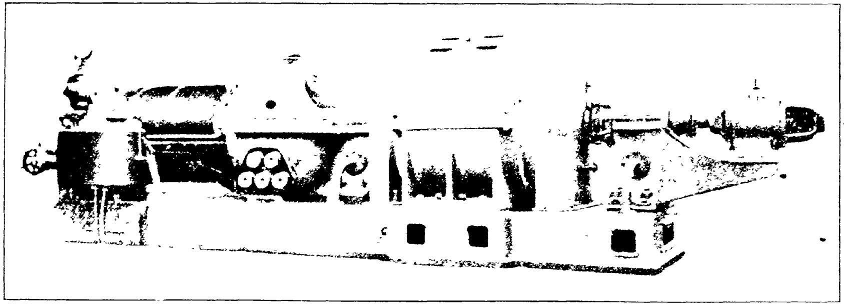


FIG. 7.—VIEW SHOWING SHAFT AND BLADES OF A BRUSH-PARSONS COMPOUND TURBINE

From Fig. 6 it will also be seen, that the steam as it flows from the steam pipe and stop valve on its way towards the working cylinders, first passes through a double-beat equilibrium valve which is controlled by a centrifugal governor. This governor, however, only comes into action in the case of emergency. After the steam has passed the emergency valve, it is then admitted to the turbine through a separate valve, controlled by its own ball governor for regulating the admission of steam according to the load. The clearance space between the bottom of this second valve and the high-pressure cylinder is as small as possible, so that the effect of the throttling of the steam



BRUSH PARSONS TURBO-ALTERNATOR.
Made by the BRUSH ELECTRICAL ENGINEERING CO., Ltd.,
Loughborough, England

by the second governor, may be as "kittle" as possible, and thus regulate the r.p.m. within 5 per cent. from full load to no load.

The Brush Company guarantee in the larger sizes a consumption of 17 lbs. steam at full load, 18 lbs. at $\frac{3}{4}$, 20 lbs. at $\frac{1}{2}$, and 27 lbs. at $\frac{1}{4}$ load per kw.-hour, with an initial pressure of 180 lbs. per square inch, a superheat of 175° F., and $27\frac{1}{2}$ inches vacuum.

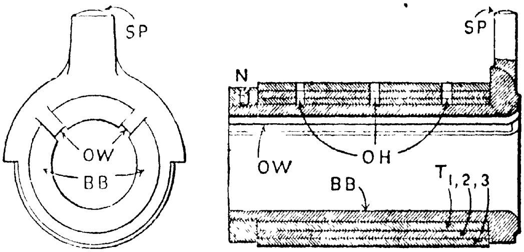


FIG. 8.—GENERAL FORM OF FLEXIBLE BEARING FOR PARSONS' TURBINE.

Bearings.—The bearings consist of a gunmetal or brass bush B B (Fig. 8), with oil ways O W, nut N, and steadying-pin S P. This bush is surrounded by several concentric brass or steel tubes T_1, T_2, T_3, fitting loosely inside each other. The lubricant enters by oil holes O H, into the interstices between the tubes, in a very thin film, which has great viscosity, and hence produces considerable resistance to rapid lateral displacement of the bearing bush. The film of oil also tends to centre the shaft in the bearing, and forms a cushion which effectually damps the vibrations arising from any errors of balance. This form of bearing is very durable, and is used for speeds over 2,000 revolutions per minute; whereas, bearings lined with white metal are more suitable for slower speeds.

Relative Spaces Required for Parsons' Turbine and Reciprocating Engines.—The accompanying Fig. 9, shows the relative spaces required for central electric station engines of these two main types. The vertical reciprocating engines in the background indicate up to 1,400 H.P., while the turbine in the foreground gives 1,500 kw., which is equivalent to an engine indicating 2,000 H.P.* The relative floor area occupied by the vertical engine plant and the turbine is $2\frac{1}{2}$ to 1. The vertical set takes about ·5 of a square foot per kw., whilst the turbine only requires ·2 of a square foot per kw. The respective heights for 3,000 H.P.

* A kilowatt is 1,000 watts, with symbol kw., and 1 kilowatt = 1·34, or roughly $1\frac{1}{3}$ horse-power.

sets are 36 feet for the vertical steam engine and only 9 feet for the turbine. This saving of space is often of great value in centres of cities, where there is difficulty in getting proper accommodation for building large central power stations.

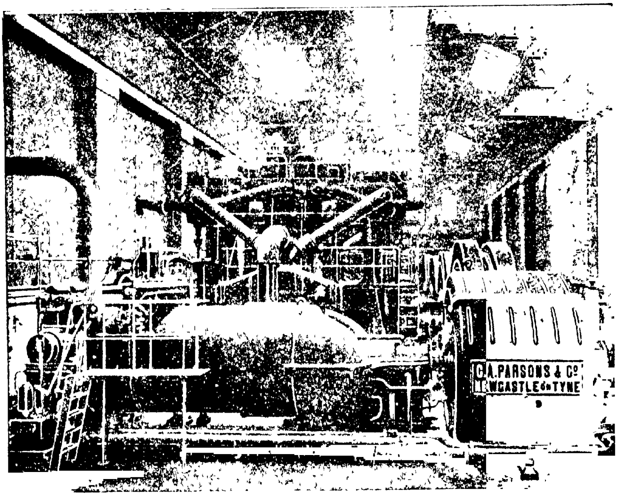


FIG. 9. LARGE 1,500 KW. TURBO ALTERNATOR.
Newcastle-upon-Tyne Electric Supply Coy., Wallsend (1902)

Superheated Steam.—The amount of clearance between the fixed and moving vanes in a Parsons turbine is very small. Consequently, when saturated steam is used, the water deposited on the fixed parts of the turbine will cause friction by coming into contact with the rotating parts. This condensation and friction may be greatly reduced by superheating the steam and jacketing the turbine. It will be seen from the results of the tests which are given afterwards, that superheating the steam improves the efficiency of a turbine. Mr. Parsons considers that 130° F. of superheat reduces the steam consumption of his turbine about 12 per cent. The results of the tests upon these turbines, show a greater percentage increase in efficiency by superheating the steam, than is accounted for by calculation and mere thermo-dynamic reasoning. It is therefore probable, that the diminution of the previously mentioned fluid friction

by superheating the steam, accounts in part for the increased economy obtained by superheating.

Effect of Vacuum on the Consumption of Steam. A very high vacuum is of the greatest importance in a turbine, since the expansion of the steam can be effected inside the turbine, right down to the vacuum pressure at the entrance to the condenser, whereas, this is not so advantageous in the case of reciprocating engines, for the following reasons:—The low-pressure cylinder would require to be very large to deal with the enormous volumes of steam, resulting in much greater cost, friction, and condensation. Also, the required size of the exhaust ports, passages, and valves would present many difficulties, and thus make the whole plant very bulky, expensive, and, perhaps, less efficient.

The following conditions are necessary for obtaining the best vacuum with turbines:—

The *first* point is to avoid all air leaks. This is easily accomplished in a turbine plant, as the only places where leakage of air is possible, are where the turbine shaft comes out of the low-pressure cylinder casing. Any leakage of air there, may be rendered very small by packing the glands with steam, so that the leakage is steam and not air.

The *second* point is to have a condenser with sufficient cooling surface. This may be attained by the most suitable arrangement of the tubes, with ample spaces between them for the steam to pass and the proper velocity of the cooling water through them. Also, a good supply of cooling water with an efficient means of cooling the condensed steam so as to keep the air-pump cool, and full provision for extracting by the air-pump the small quantity of air which must leak into the turbine.

In the *third* place, the condenser should be placed straight underneath, and close to, the turbine, so that the drop in the vacuum may be as small as possible between them.

The following table shows the effect of variation of vacuum on the consumption of steam in the case of a 1,500 kw. set. For smaller sizes the variation is not so marked:—

Vacuum bar = 30 ins. Inches	Difference in steam consumption per 1 in. of vacuum
29	6 per cent
28	5 ,,
27	4 ,,
26	3½ ,,
24	3 ,,
22	2¼ ,,
16	2 ,,

By paying special attention to the above requirements, it has been found unnecessary to use a much larger condensing plant than has hitherto been done for the same power with reciprocating engines. In the case of the most recent condensers for steam turbines from 10 to 12 lbs of steam are condensed per square foot per hour; and at this rate of condensation, vacua of from 27·5 to 28 inches, with barometer at 30 inches, can be obtained at full load. The amount of cooling water generally allowed is about fifty times the full-load steam consumption, which will increase the vacuum under normal conditions by from ½ to 1 inch over that obtained by the usual rule of twenty-five to thirty times the amount of steam used.

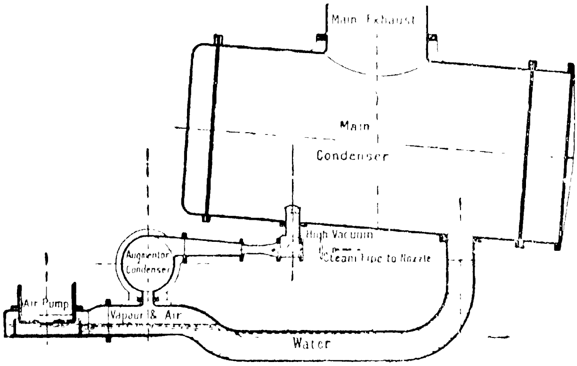


FIG. 10.—ARRANGEMENT OF PARSONS' VACUUM AUGMENTOR.

The Vacuum Augmentor.—With regard to extracting the air, a great improvement has been effected by the recently-introduced vacuum augmentor. In it, the air-pumps are placed about 3 feet below the bottom of the condenser. From any convenient part of the condenser, preferably near the bottom, a pipe is led to an auxiliary condenser, the size of which is about $\frac{1}{20}$ the cooling surface of the main condenser. In a contracted portion of this pipe a small steam jet is placed, which acts in the same way as a steam exhaustor, or the jet in the funnel of a locomotive. It sucks nearly all the residual air and vapour from the condenser and delivers it to the air-pumps. A water seal is provided, as shown on Fig 10, to prevent the air and vapour returning to the condenser. Thus, if there is a vacuum of 27·5 inches to 28 inches in the condenser, there may be only about

26 inches in the air-pump, which need therefore only be of small size. The steam jet compresses the air and vapour from the condenser to about half its original volume. The small quantity of steam from this steam jet (which is only about $1\frac{1}{2}$ per cent. of that used by the turbine at full load), together with the air extracted, are cooled down and condensed by the auxiliary condenser, which is generally supplied with cooling-water in parallel with the main condenser. In this connection it should be observed, that condensation in a condenser takes place much more rapidly and effectually if the air is thoroughly extracted, than if there is much air present, as the air seems to form a blanket or non-conductor round the tubes and retards the steam getting to them.

Tests of Parsons' Turbines.—In order to give the student an idea of the efficiency of these turbines when coupled direct to electric generators, three sizes have been chosen—viz., 200, 500, and 1,500 kw. respectively.

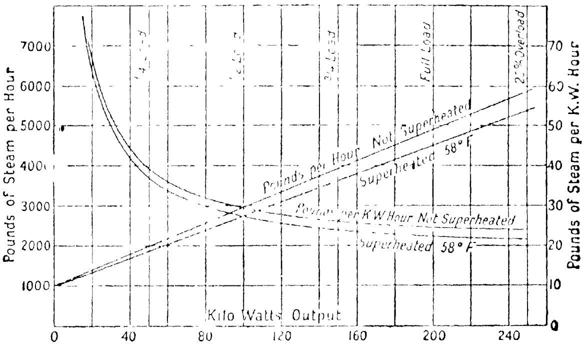


FIG. 11.—PLOTTED RESULTS OF TESTS OF A 200 KILOWATT PARSONS' TURBO GENERATOR.

First, Test of a 200-kw. Turbo Generator for Continuous Currents.—The several curves plotted on Fig. 11 were derived from tests taken at $\frac{1}{4}$, $\frac{1}{2}$, $\frac{3}{4}$, full, and 25 per cent. over loads, with steam of 110 lbs. boiler pressure. They show the pounds of steam used per hour, as well as the pounds of steam per kw.-hour, without and with a very mild superheat of 58° F. at those loads. The student should study these curves, and thus be prepared to plot down the results enumerated in the two following tables, for the 500 and 1,500 kw. turbo-alternators.

Second, Test of a 500-Kw. Turbo-Alternator.—In this case the steam pressure was also 140 lbs. by the boiler gauge, but no superheat was applied.

The tests were specially made to show the effects of the vacuum on the consumption of steam. In the following table we have full load, half load, quarter load, and no load steam consumption per kw.-hour for every inch of vacuum from 22 inches to 28 inches. At 28 inches vacuum the full load consumption is only 22·2 lbs. per kw.-hour, and with the same load but a 22-inch vacuum the consumption is 28·9 lbs., or an increase of 31 per cent. At quarter load with 28 inches vacuum the consumption is 32·4 lbs., and with 22 inches vacuum 46·3 lbs., or an increase of 43 per cent. These figures prove, conclusively, the enormous gain obtainable by having a good vacuum. It is calculated that on full load there is a gain of at least 4 per cent. in steam consumption for every extra inch above 25 inches, and on lighter loads the gain is still greater with this size and type of plant.

Steam Consumption of a 500 - Kw. Parsons' Turbo - Alternator running at 2,500 Revolutions per Minute with 140 Lbs. Steam Pressure at the Stop Valve, and no Superheat.

Vacuum Constant from Full Load to No Load.	Steam Consumption per Kw.-hour without Superheat.			
Inches of Mercury.	Full Load	½	¼	No Load.
29	...	...	...	1,500
28	22·2	25·6	32·4	1,700
27	23·1	26 9	34·5	1,900
26	24·0	28·2	36·6	2,100
25	25·1	29 7	39·0	2,300
24	26·2	31·2	41·2	2,500
23	27·5	32·9	44·8	2,700
22	28·9	34·7	46·3	2,900

Third, Test of a 1,500-Kw. Turbo-Alternator.—In the following table and Fig. 12 are given results and curves of a test on a 1,500-kw. plant for the Sheffield Electric Light Central Station. These show the consumption of steam under varying loads, with and without the vacuum augmentor. The steam used by the augmentor is included, and amounted to 450 lbs. per hour. The difference in vacuum is also shown, and when it is remembered that the augmentor jet only took about 1½ per cent. of the full-

load steam consumption, it is easily seen from the gain of vacuum where the total gain comes in by the use of the vacuum augmentor. In this case the vacuum was not as good as it should have been, as the cooling water was 85° F., and was only about thirty times the weight of steam used at full load.

STEAM CONSUMPTION OF A 1,500-KW. PARSONS' TURBO-ALTERNATOR WITH VACUUM AUGMENTOR

Pressure in Lbs per Sq. Inch above Atmosphere.	Superheat in Degrees Fahr.	Vacuum Inches	Revs per Minute.	Load in Kw.	Lbs of Steam used per Hour	Lbs of Steam used per Kw hour
113·6	108·3	26·69	1,455	1,316·5	24,732	18·76
111·6	156·4	27 12	1,500	1,061·6	19,830	18·66
141	113	27·72	1,500	512 7	11,425	22 3
154	47 5	27 72	1,500	.	3,128	. .
WITHOUT VACUUM AUGMENTOR						
115·6	143	25·18	1,500	1,029 3	21,264	20 7
137	119	25·97	1,500	534·25	12,820	24 02
150·3	72·4	26·62	1,500	...	2,957·4	...

In all the above tests the barometer is taken as 30 inches.

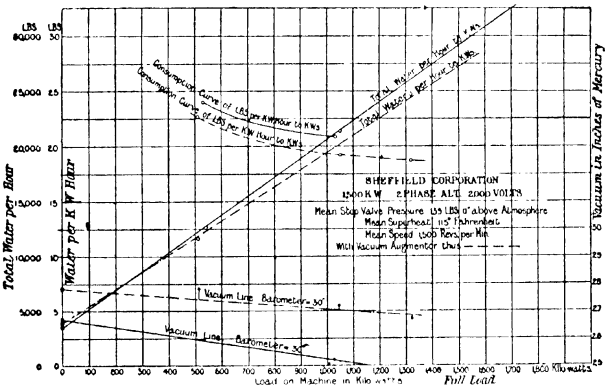


FIG. 12.—PLOTTED RESULTS OF TESTS OF A 1 500-KW PARSONS' TURBO-ALTERNATOR TO SHOW THE EFFECT OF THE VACUUM AUGMENTOR.

Marine Turbines.*—In January, 1894, Mr. Parsons formed a separate company for the purpose of developing his turbine for marine propulsion. The company set about building a boat of practical size at Wallsend-on-Tyne, and the first preliminary trial of it was made on November 14, 1894. It was this same yacht, "Turbinia," which created so much astonishment and excitement, by steaming in and out amongst the men-of-war at the Jubilee Naval Review in 1897, at the then extraordinary speed of 34 knots.

To give the student a good general idea of this interesting, instructive, and successful application, we cannot do better than reproduce here some of the figures specially presented for this book by the "Parsons' Marine Steam Turbine Company," together with a short abstract from Prof. Ewing's report upon the vessel.

General Description of the "Turbinia" and her Engines.†—The "Turbinia" is 100 feet in length, of 9 feet beam, and had a displacement under the trial conditions of about 44½ tons. She has three screw shafts, each of which is directly driven by one of Parsons' steam turbines. The turbines are of the parallel flow type, the general course of the steam through them being parallel to the axis of rotation. In each of the three turbines steam enters at the forward end, and streams aft through an annular space formed between the outside of a cylindrical boss, which is carried by the shaft, and the inside of a corresponding cylindrical casing, in which the turbine is enclosed.

The three turbines form a compound series. Steam being first admitted to the turbine E, on the starboard side, it passes then to the turbine G, on the port side, and lastly, to the low-pressure turbine I, which is placed amidships. The speed is controlled by a regulator valve D, on the admission pipe to the high-pressure turbine E. When running at full power the admission pressure is about 155 lbs by gauge, or 170 lbs. absolute, and this was reduced by expansion in the turbine to about 1 lb. per square inch before the steam was discharged to the condensers, L. The position of the turbines and the general arrangement of the machinery are shown in the accompanying figures. The full-page illustration (Fig. 13) shows a midship section of the boat from the after end of the boiler to the stern, also a corresponding sectional plan. The left-hand view (Fig. 14) is a transverse section through the forward end of the engine-room, looking aft. The right-hand view (Fig. 15) is a section showing the high-pressure, intermediate, and low-pressure turbines. The left-hand view (Fig. 16) is a transverse section through the forward stokehold. The right-hand view (Fig. 17) is a section at the after end of the vessel showing the propellers.

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* See "The Steam Turbine and its Application to the Propulsion of Vessels," by The Hon. C. H. Parsons, M.A., F.R.S., *Proc. I.N.A.*, 1903.

† In the full-page illustration (Fig. 13) it was found necessary to bring the boiler A, closer to the engines than in the actual boat, and to show the steam pipe C as cut in order to illustrate these parts within the limits of the page —A.J.

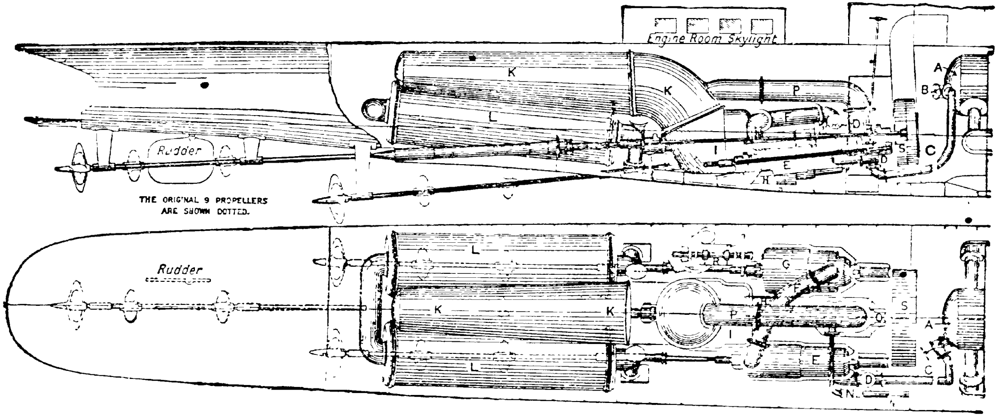


FIG. 13.—SECTIONAL ELEVATION AND PLAN OF THE "TURBINIA'S" ENGINES, &C., FROM THE FORWARD STOKEHOLD TO THE STERN SCREWS.

INDEX OF PARTS TO FIGS. 13 TO 17

A for Boiler.
B ,, Boiler stop valve.
C ,, Main steam pipe.
D ,, Regulator valve.
E ,, H.P. turbine.
F ,, H.P. turbine exhaust to I.P. turbine.
G for I.P. turbine.
H ,, I.P. turbine exhaust to L.P turbine.
I ,, L.P. turbine.
K ,, L P. turbine exhaust to condensers
L, L ,, Condensers.
M for Main steam to astern turbine.
N ,, Astern turbine regulator valve.
O ,, Astern turbine
P ,, Astern turbine exhaust to condensers L
R ,, Air pump.
S ,, Fan for boiler fires.

From the low-pressure turbine I, steam is taken by a large exhaust pipe K, of unusual shape and cross area to a pair of cylindrical surface condensers L, with a total cooling surface of 4,200 square feet, through the tubes of which a circulation of sea water is kept up by means of hinged scoops projecting from the sides of the boat. The scoops may be set to make water enter at either side and pass out at the other, so that any

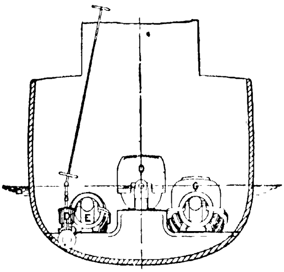


Fig. 14.—Transverse Section through the Forward End of the Engine - Room (Looking Aft).

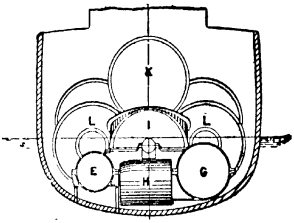


Fig. 15.—Cross Sectional View, showing the High, Intermediate and Low - Pressure Turbines (Looking Aft).

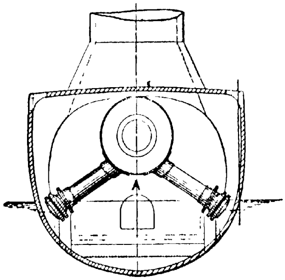


Fig. 16.—Transverse Section through the Forward Stoke-hold, showing the Boiler.

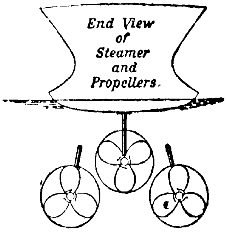


Fig 17.—View at the After End of the Vessel, showing the Propellers.

obstruction of the tubes (which are $\frac{5}{16}$-inch internal diameter) to the flow may be cleared by reversing the direction of the stream. The air-pump R, is a small reciprocating pump, which appears in the plan on the port side. It is driven by gearing from the screw shaft of turbine G.

The turbine shafts slope down towards the stern, the middle shaft with an inclination of about 1 in 16, and the two side shafts with an inclination

of about 1 in 8½. Their diameter is 2½ inches. There are three propellers on each shaft, making nine in all, which are distributed as shown in the plan. The screws are 18 inches in diameter and 2 feet in pitch. When running at full speed they make about 2,200 revolutions per minute.

The middle shaft projects forward through the low-pressure turbine I, and has mounted on it a fan S, for forcing the draught. Air is delivered by this fan into the after stokehold, and passes to the forward stokehold, round the sides of the boiler A The air pressure at full speed is about 7 inches by water gauge. The boiler A, is a double ended one, and has a heating surface of 1,100 square feet and 42 square feet of grate surface; is of the straight tube type, the tubes being ½ inch internal diameter and $\frac{1}{16}$ inch thickness; two return water tubes of 7-inch diameter at each end connect the top drum with the water pockets. The diameter of the top drum is 34 inches, and sufficient to enable a tube to be drawn and replaced by a workman in the top drum. Steam is supplied at the working pressure of 210 lbs. by gauge through the stop valve B, steam pipe C, and regulator valve D, to the high-pressure turbine E.

The auxiliary machinery consists, in addition to the air pump, of a small spare air pump, small circulating pump for use when the scoops are not available, main and spare feed pumps, bilge ejector, and oil pump for maintaining a circulation of oil through the bearings of the turbine shafts and through the thrust blocks. The total weight of the turbine engines is 3 tons 13 cwts.

For the purpose of reversing the motion, a separate turbine O, is provided between the low-pressure turbine I, and the fan S. When the boat is going ahead, this reversing turbine O, being permanently mounted on the central propeller shaft, will revolve, but its casing will be kept connected to the condensers, and the amount of work spent in turning the turbine shaft will be insignificant.

The mechanical friction of the turbines is particularly small, and the work spent on friction is not materially increased by increasing the range of expansion. This allows the steam to be profitably expanded much farther than would be useful or even practicable in an engine of the ordinary kind. Apart from questions of friction, the addition of weight and bulk to allow for this extended expansion would be enormous in the ordinary engine. In the turbine it is very moderate. Steam is expanded nearly two hundredfold in the "Turbinia," and this is accomplished with engines which are much lighter than reciprocating engines of the same power, although in these the expansion would be much less complete. Taking the propulsive coefficient as ·5, then the steam in the full-power trials was doing work in the turbines at the rate of about 2,100 H.P., and the consumption of the steam was barely 14½ lbs. per H.P.-hour.

The Advantages of One Propeller on Each Shaft.—In May, 1902, trials were made with only one propeller on each propeller-shaft, instead of the three which had been originally fitted to each shaft. These propellers were each 28 inches diameter and 28 inches pitch. The results of the single propellers show the greatest advantage at about 21 knots, where the gain in speed over the former arrangement amounted to 2 knots. The loss of efficiency which had been observed at certain speeds in some vessels fitted with tandem propellers on each shaft, seems to be due to "*interference*" in the proper flow of the water.

Turbine-driven Boats for Commercial Purposes—"King Edward."

History, Sizes, Turbines, and Propellers.—The first passenger vessel to be propelled by steam turbines was the "King Edward." This boat was built in the spring of 1901 by Messrs. W. Denny & Bros., of Dumbarton, and engined by the Parsons Marine Steam Turbine Company, Ltd., to the order of Captain John Williamson, of Glasgow. The length is 250 feet, the beam 30 feet, with about 6 feet draught of water. The turbines are similar in construction to those of the "Turbinia," and consist of three turbines—viz., one high pressure driving the centre shaft and two of low pressure working in parallel for driving the side shafts. In the exhaust casing common to the low-pressure turbines is placed the reversing turbine. On the centre shaft there is one propeller of 57 inches diameter, and on each of the side shafts are two propellers of 40 inches diameter at about 9 feet apart

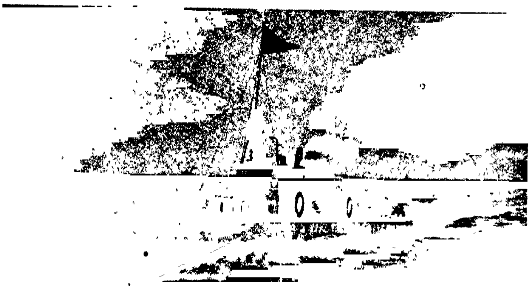
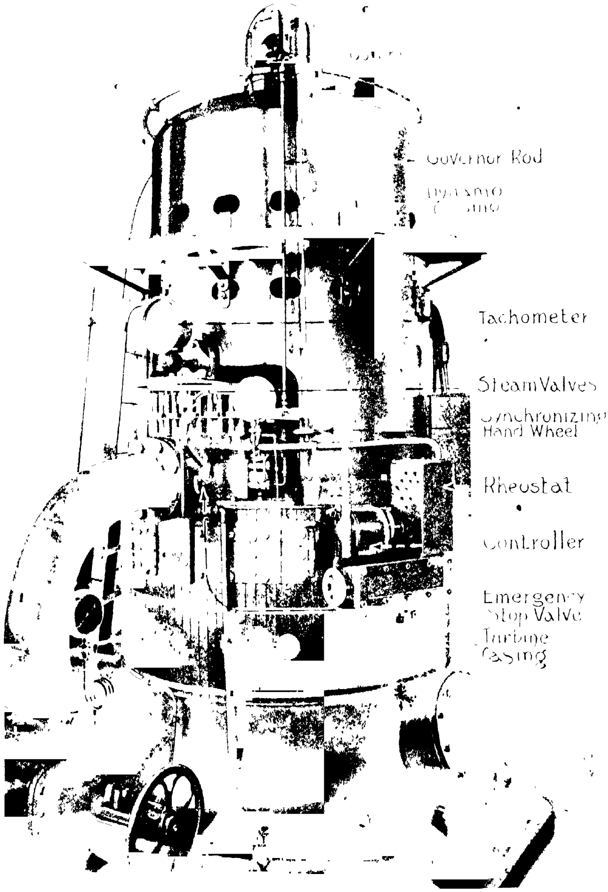


FIG. 18.—S.Y. "TURBINIA" AT FULL SPEED.

Going Ahead, Manœuvring, and Going Astern.—When the main stop valve is opened to the high-pressure turbine, all the turbines go ahead. When it is desired to turn or manœuvre the vessel by aid of the propellers, then the central turbine and its propeller remain idle, but steam may be admitted alternately and directly into either of the side low-pressure turbines—or into either of the side turbines going ahead, and at the same time into the reversing or astern-going one. When merely going astern, the reversing turbine alone is employed, whilst the others are idle.

Auxiliary Machinery and Boiler.—The auxiliary machinery is of the usual type, and needs no special mention, beyond stating that the air pumps are worked by worm wheels from the low-pressure turbine shafts. The boiler is of the usual return-tube and double-ended type, with a working steam pressure of 150 lbs. per square inch.



The 500 K. W. Curtis Steam Turbo-Generator.
Made by THE BRITISH THOMSON-HOUSTON CO., LTD., Rugby.

Trial and Speed.—The trial of the "King Edward" was made on June 26, 1901, in the Clyde, along the Skelmorlie mile, when a mean speed of 20·48 knots was recorded, the revolutions of the centre shaft being 505, and the side shafts 755.

Power and Coal Consumption.—The indicated horse-power was estimated to be 3,500 from model experiments in the tank at Dumbarton. The average sea speed on the run of about 160 miles to Campbeltown and back in the 1901 season was 19 knots, and the average coal consumption, including lighting up, &c., was 18 tons per day, or 1·8 lbs. per equivalent indicated horse-power-hour.

The Curtis Turbine.—This turbine was first made by Mr. C. G. Curtis, of New York, in 1896. It is manufactured by the General Electric Co., U.S.A., as well as by the British Thomson-Houston Co., of Rugby, to whom and to *The Tramway and Railway World*, of London, we are indebted for the following illustrations and data. As will be seen from the accompanying Plate, showing the outside view, and Fig. 19, this turbine differs in outward form from the previous kinds. The dynamo is placed immediately over the turbine, and thus the whole makes a very compact plant by occupying a minimum of floor space. It avoids the very high speeds of the De Laval and does not possess the very great number of vanes of the Parsons turbine. The torque is a maximum when the turbine wheel is at a standstill, and would become zero if the velocity of the periphery could attain the velocity of the steam jets. Hence, a maximum output as well as the best economy are obtained when the torque is one-half the maximum, where the velocity of the periphery is about one-half the natural velocity of the steam. It was shown in the description of the De Laval turbine, that 1 lb. of dry saturated steam, when expanding from a pressure of 175 lbs. absolute down to atmospheric pressure, acquires a velocity of about 2,900 feet per second; whereas, if we carry this expansion down to 1 lb. absolute, then the steam acquires a velocity of about 4,000 feet per second.

Now, since the kinetic energy, $E_K = \dfrac{Wv^2}{2g}$, we get, in the first case, 130,590 foot-lbs. per lb. of steam used, and, in the second case, 248,447 foot-lbs. per lb. of steam; from which, it will be seen, that only about one-half of the available energy of the steam is usefully applied when expanding down to atmospheric pressure. But 1 kw.-hour is equal to 2,660,000 foot-lbs. Therefore, by calculation, for the second case (where the steam expands to 1 lb. absolute) only about $10\frac{1}{2}$ lbs. of steam are required per kw.-hour. Consequently, if the turbine actually requires 20 lbs. of steam per kw.-hour with a 28-inch vacuum, whilst running under the latter conditions, we obtain 53 per cent. of the available energy in the steam as useful work.

TABLE OF THE CHIEF MEASUREMENTS AND SPEEDS OF RECENT ATLANTIC LINERS.

Vessel	"Campania"	"Oceanic"	"Deutschland"	"Kaiser Wilhelm II"
Built,	1893	1899	1900	1902
Owners,	Cunard Co.	White Star Co.	Hamburg American	North German Lloyd
Length (extreme)	622 ft 0 ins.	704 ft. 0 ins	684 ft 0 ins	706 ft 6 ins
Length (B.P.),	600 ft 0 ins	685 ft 0 ins	662 ft 9 ins	
Breadth,	65 ft 3 ins	68 ft 0 ins	67 ft 0 ins	72 ft. 0 ins
Depth,	41 ft 6 ins	49 ft. 6 ins	44 ft 0 ins	52 ft 6 ins
Gross tonnage,	12,500	17,274	16,502	20,000
Load draft,	25 ft 0 ins	32 ft 6 ins	29 ft 0 ins	29 ft 0 ins
Displacement,	18,000 tons.	28,500 tons	23,620 tons.	26,000 tons.
	1st 2nd 3rd	1st 2nd 3rd	1st 2nd 3rd	1st 2nd 3rd
Passengers,	600 400 700	410 300 1,000	693 302 288	775 343 770
Boilers,	1 S E. and 12 D.E.	15 D E.	4 S E. and 12 D E	7 S.E. and 12 D E
Heating surface,	82 000 sq. ft.	74,686 sq ft	85,468 sq ft.	107,643 sq. ft.
Fire grate,	2,630 sq ft	1,962 sq ft.	2,188 sq. ft	3,121 sq. ft
Working pressure,	165 lbs	192 lbs	220 lbs	225 lbs.
Type of engine,	Triple	Triple.	Quadruple.	Quadruple
Sets,	2	2.	2	4.
Cylinders diameters,	2 of 37″ × 79″ × 2 of 98″	47½″ × 79″ × 2 of 93″	2 of 36·6″ × 73 6″ × 103 9″ × 2 of 106·3″	37 4″ × 49 2″ × 74·8″ × 112 2″
Stroke,	69 ins	72 ins	72·8 ins	70 8 ins.
Speed,	22 knots	20 knots	23½ knots	23½ to 24 knots
I H.P.	30 000	27,000	36,000	38,000 to 40 000

This table is an abstract of the more complete table given by Dr. Robert Caird in his recent lecture on "Developments in the Means of Communication by Sea during the Nineteenth Century," as delivered by him on 6th April, 1904, before the Royal Philosophical Society of Glasgow Since then, both the turbine steamers "Lusitania and the "Mauretania" have exceeded 25 knots.

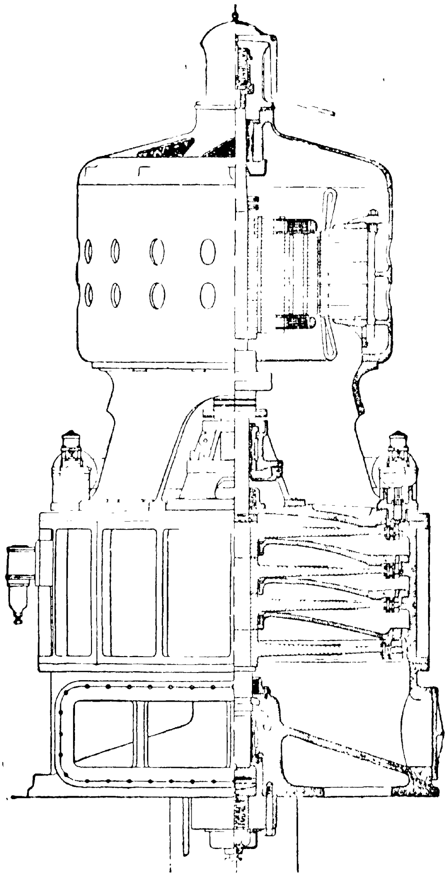


FIG. 19.—HALF OUTSIDE AND HALF SECTIONAL ELEVATION OF THE CURTIS STEAM TURBINE AND DYNAMO, ILLUSTRATED BY THE ACCOMPANYING PLATE.

From actual test recently made of a 500 kw. Curtis turbine at Cork, it was found that the steam consumpt was 20.5 lbs. per kw.-hour, using steam at 150 lbs. per square inch, superheated 100° F. and with a 27-inch vacuum. But, when the vacuum was improved to 28½ inches, and the superheat raised to 150° F., the steam consumpt fell to 19 lbs per kw.-hour, which is equivalent to 14 lbs of steam per H.P.-hour.

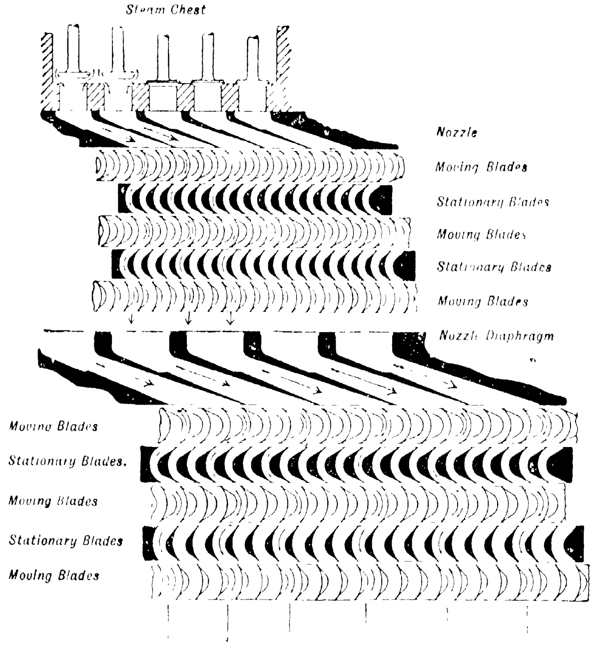


FIG. 20.—DIAGRAM OF NOZZLES AND BUCKETS IN THE CURTIS STEAM TURBINE.

General Description of a 500 Kw. Curtis Turbine.—The 500-kw. turbine (Figs. 19 and 20) has three wheels or rows of movable buckets in each of the two stages of the steam expansion. Each wheel consists of a solid steel disc with the buckets cut from the solid rim, and the three wheels are bolted to a common hub which fits the shaft. The total number of buckets on the wheels in this turbine is only 1,395, which is but a small number compared with that of a Parsons turbine giving the same output.

The stationary buckets have approximately the same shape as the wheel buckets, and are fixed to the casing of the turbine The number of stationary buckets is dependent on the number of nozzles in the stage, as they need only cover the same periphery of the wheels as the nozzles.

On account of the large increase in volume of the steam and to keep the depth of the buckets as small as possible, the wheels in the last stage of expansion are completely surrounded by nozzles and stationary buckets.

In the most recent designs of this turbine, the construction of the wheels has been slightly modified from that indicated above.* Instead of using separate discs bolted together, a single cast-steel wheel is used with the bucket rings or segments bolted to the sides at the periphery of the wheel.

Nozzles and Buckets.—Steam is admitted through a number of nozzles which are slightly conical In these nozzles the steam partially expands and acquires a velocity of about 2,000 feet per second in a four-stage turbine, thus converting part of its potential energy into kinetic energy due to its expansion After leaving the nozzles, the steam passes successively through two or more lines of buckets, as shown by Fig. 20. Steam is directed against the first set of movable buckets, and then re-directed against the second moving set of buckets by passing through the first set of the fixed buckets, and so on, until it enters the next stage, where the same sequence of events take place, until the steam is finally brought to rest in the condenser.

By this means a high steam velocity is made to efficiently impart motion to a comparatively slow-moving wheel

The nozzle is generally made up of many sections or units adjacent to each other, so that the steam passes to the buckets in a broad sheet when all the sections are open

This process of expansion in the nozzle, and subsequent abstraction of the velocity of the steam by successive impacts of gradually diminishing force upon the buckets is repeated two or more times, each such complete repetition being designated as a stage There are various numbers of stages and various numbers of lines of buckets in each stage. The number of stages and the number of lines of buckets in each stage are governed by the degree of expansion, the desirable or practicable peripheral velocity of the buckets, and by various conditions of mechanical expediency.

A steam-tight diaphragm is placed between each stage The only outlet for the steam to pass into the next stage is through the nozzles in the said diaphragm.

The leakages in the buckets are, therefore, not total losses except in the last stage, since the steam has to pass through the nozzles in the succeeding diaphragm and thus do still further work.

Clearance between Stationary and Rotating Blades.—The clearances between stationary and rotating parts vary from ·02 inch in the small to ·08 inch in the 5,000 kw. turbine. These are the clearances between the outer shrouding of the buckets and the rings of metal from which the buckets are cut. The buckets are made a little narrower than the rings from which they are cut, so that they should not come into contact with each other, even if the rings touch each other.

The adjustment of the clearance is made by means of the heavy screw bolt shown at the bottom of the step bearing, which supports the footstep blocks (Fig. 19). Inspection holes are provided in the casing, so that the clearances between the buckets may be observed from time to time.

Centrifugal Governor.—The governor is of the centrifugal spring loaded type, and is generally set for a speed regulation of 2 per cent. between full and no load, with a maximum momentary variation of 4 per

cent. For the purpose of synchronising, and to make the turbine take its proportional share of the load, a supplementary spring is introduced, which acts in conjunction with the main governor spring, and allows the speed of the turbine to be varied $2\frac{1}{2}$ per cent on each side of the normal speed.

In the smaller turbines, this adjustment is made by a hand-wheel placed in a convenient position near to the main throttle valve. In the larger turbines, from 1,500 kw. and upwards, a small motor actuates this spring. The motor is controlled by a double pole, double throw switch, placed on the switchboard, or at any other convenient position. The centrifugal governor, which is situated at the extreme top of the vertical shaft, operates a controller for opening or closing the electrical circuits of the magnets, which actuate the pilot valves. These pilot valves in turn admit or exhaust steam from the spaces behind the pistons of the main valves, one of which is provided for each nozzle or group of nozzles. The governor thus opens and closes these valves, and varies the number of nozzles in service in proportion to the load. Since each nozzle represents a comparatively large proportion of the whole power at light loads, one of these valves would be called upon to open and close alternately in rapid succession, in order to hold the speed constant. To avoid this, a balanced throttle valve is placed in series in the steam path of the first valve and corresponding nozzle, and so connected to the centrifugal governor that it must be fully opened before the governor can open the next valve, and again fully closed before the controller moves backwards and closes the valve last opened. The result is, that the turbine can run at a constant speed on any load, although only a small portion of the steam will be throttled. Thus, the following disadvantages are avoided:—(1) Admitting steam to the buckets in puffs, for these puffs tend to create a variation in speed at light loads, and to make the parallel running of the generators difficult; and (2) of throttling the whole steam flow.

The nozzles in the later stages are also, in some of the turbines, opened and closed by the governor so as to maintain an adjustment of pressures proportional to that of the first stage, and thus still further tend to steady running, under wide and quick ranges of load.

A sufficient number of nozzles are provided in the first stage to run the turbine non-condensing at full load. Hence the turbine has an overload output of about 100 per cent. when condensing with a 28-inch vacuum.

Emergency Governor.—In addition to the speed regulating governor, an emergency governor is provided, whose function is to trip a trigger, should the speed of the turbine increase 15 per cent. above the normal. The tripping action permits a weight to fall and give a hammer blow instantaneously to a butterfly or damper valve in the main steam pipe, thus closing the valve. This emergency governor is located on the shaft between the turbine and the generator, and consists of an ordinary centrifugal device balanced against a spring. The governor automatically returns itself to normal position when the speed is reduced, so that it is not necessary to stop the turbine, but the butterfly valve and weight must be brought to their starting position by hand. Such a precautionary device is rarely brought into action, but experience with large engines has shown it to be desirable.

Vertical Shaft, Footstep Bearing and Oiling Arrangement.—One of the most noticeable features in the Curtis turbine is the vertical shaft. This type of construction has been adopted for all sizes from 500 kw. upwards.

The use of a vertical shaft necessitates a footstep bearing, which is fixed

to the bottom of the base, and removable from below through a pit or opening, provided in the foundation for this purpose. The bearing part proper, which supports the rotating elements of the turbine and generator, consists of two circular cast-iron plates, one fixed to and rotating with the shaft, the other fixed to the base. Oil is forced between these plates through a hole in the stationary plate, from the centre outwards in a thin film, separating the two plates by about ·005 of an inch. From the

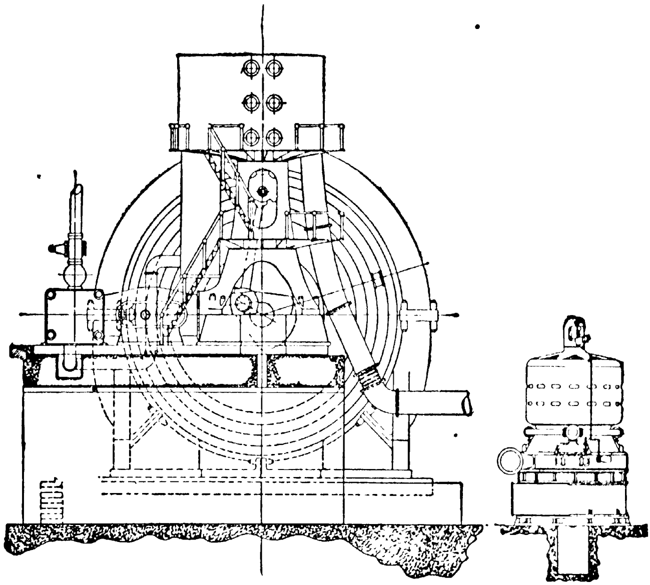


FIG. 21.—SHOWING COMPARATIVE SIZES OF A 5,000-KW 75-REVOLUTIONS-PER-MINUTE CORLISS ENGINE WITH GENERATOR, AND OF A 5,000-KW. 500-REVOLUTIONS-PER-MINUTE CURTIS TURBINE WITH GENERATOR.

footstep bearing the oil passes upwards, and lubricates the guide bearing which is placed immediately above it. The oil is forced between the plates by a pressure pump, designed to give approximately a constant pressure and a constant flow.* The pressure required depends upon the weight of the rotating part and the area of the bearing blocks, and in a small degree upon the temperature of the oil. The side thrust of the steam jet on the

* See the author's *Text-Book of Applied Mechanics and Mechanical Engineering*, vol. i., Lecture VI., "Methods of Lubricating Footstep Bearings," &c.

buckets is not noticeable, as the oil pressure required is exactly the same, whether the turbine is standing still or rotating. The oil pressure in the step bearing, varies from 175 lbs. per square inch in the 500-kw. to about 900 lbs. per square inch in the 5,000-kw. turbine. Experiments have demonstrated, that the safe minimum amount of oil required for the bearings, is from half-a-gallon for the 500-kw. to four gallons per minute for the 5,000-kw. turbine. A small amount of this oil is shunted through a resistance, and delivered to a tank placed above the level of the turbine The oil flows by gravity from this tank to the upper and middle bearings, and a sight-feed is introduced for each branch The oil from all three bearings is returned to one common tank, where it is strained and cooled before being again put in circulation.

To prevent the turbine being run either by steam or through the generator acting as a motor before oil begins to flow in the step bearing, or, in case the flow should cease while the turbine is working, an automatic oil flow switch is placed in the high-pressure oil supply pipe. This switch is actuated by the oil pressure. It opens and closes the electric circuit for the valve magnets, in response to the change of pressures in the pipe. If the flow of oil in the bearing falls below the pre-determined quantity or pressure, the switch opens the valve circuit, and all the valves close, and thereby immediately shut off the steam from the turbine When the switch opens, a special relay is closed, which trips the generator circuit-breaker and prevents the generator from running as a motor and driving the turbine.

Bearings.—The upper and middle bearings of the vertical shaft have no load to carry save that due to want of balance, and to any unbalanced magnetic pull in the generator. All the bearings can be taken out without disturbing the turbine or generator. The footstep blocks and the guide bearing are lowered into the pit below, the middle bearing is split in halves, and can be removed sideways, whilst the upper bearing is lifted out. As the step bearing works under atmospheric pressure, it is necessary to provide packing rings round the shaft at this point, to prevent air entering the exhaust base when the turbine is working under a vacuum, or steam escaping, when the pressure of the steam within the turbine shell is higher than the atmospheric pressure For the same reason packing is provided where the upper end of the shaft passes through the turbine cover.

It will thus be seen that special attention has been given by the designers of this turbine to ensure the sweet, uniform, continuous working of every part of this prime motor.

Efficiency of Turbines.*—It is essential that the word efficiency should be clearly defined, in order to avoid any misconception as to the figures which are given below Prof Rateau uses the expression, "theoretical consumption of the perfect machine," to denote the maximum work which the steam is capable of supplying when starting from the saturated or superheated condition in which this steam is delivered to the engine, and expanding adiabatically, with no loss of admission pressure P, to exhaust pressure p. By comparing the actual consumption of steam as measured during the tests of the machine with this theoretical consumption of the

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* Prof. A. Rateau, of Paris, in his 1904 paper at the Chicago joint meeting of the A.S.M.E. and I.M.E , of London, stated the above formulæ, to which the author has added a few remarks about the value of the French H.P. Students should try how far the formula for steam consumpt in British H.P. agrees with those previously given in this lecture.

perfect machine for identical conditions of pressure and similar states of the steam, the net efficiency of the machine is obtained. After special study of the question, Prof. Rateau has been able to draw a curve of theoretical consumption and to derive from it the following empirical formula for use when the steam is saturated and dry at admission :—

$$\text{Steam consumpt} = 0.85 + \frac{6.95 - 0.92 \log P}{\log P - \log p}.$$

This formula gives the consumption in kilogrammes per H.P.-hour as a function of the absolute pressures P and p, expressed in kilogrammes per square cm. The H.P. here used is the French "*Force de Cheval*" = 75 kg. m/s, or 75 kilogramme-metres per second = 736 watts = 542·48 ft.-lbs. per second = ·9863 British horse-power.

In British measures this formula becomes—

$$\text{Steam consumpt} = 2{\cdot}13 + \frac{16.20 - 2.05 \log P}{\log P - \log p},$$

where the steam consumption is in lbs. per H.P.-hour, with P and p in lbs. per square inch. If the steam admitted to the turbines is superheated, then, of course, in estimating the theoretical consumption, the extra calorific energy, corresponding to the superheat, must be taken into account.

Advantages and Chief Features of Steam Turbines.

1. No cylinder lubrication is required, and no "travelling oil cans" are necessary. Consequently, as the condensed steam is perfectly clean and free from oil, it can be pumped back into the boiler without any purification, thus saving the costly oil separating plant, which is usually fitted to large installations for purifying purposes.

2. The reciprocating engine is now very nearly as perfect as it is likely to be, both from a mechanical and thermal point of view.

3. The dryest and highest practicable pressure of steam, with any amount of superheat, may be used if clearance and proper material to withstand erosion be adopted in the construction of the turbine.

4. They use no more steam than the highest grade reciprocating engine.

5. The expansion of the steam can be carried to its extreme limit much more conveniently in turbines than in reciprocating engines.

6. As there are no reciprocating motions inside the turbine, all vibrations and noise are minimised.

7. On account of the practical absence of vibration, neither special foundations nor holding down bolts are required.

8. Overloading can be indulged in to a large extent.

9. They give a uniform turning moment.

10. Being the simplest form of engine, they require considerably less attendance and occupy one-fifth to one-tenth the cubic space of a reciprocating engine.

11. Steam turbines should be cheaper to build than reciprocating steam engines of equal horse-power.

12. Owing to their capability of producing great power at high speed, steam turbines coupled direct to ventilators and centrifugal pumps exhibit very good results.

LECTURE XVI.—QUESTIONS.

1. Deduce in full detail from the first principles of thermodynamics a formula, whereby the heat units due in the shape of work done by each lb. of steam may be accurately calculated. Then, make any permissible modifications upon your formula in order to simplify it for easier computation, and state clearly the assumptions upon which your modifications are based. Test your formula by ascertaining the velocity and kinetic energy per lb. of dry steam entering a De Laval nozzle at 115 lbs absolute, and leaving the wheel under a 25 and a 28-inch vacuum.

2. Find the velocity of the steam issuing from the conical nozzle of a De Laval turbine when the initial pressure of dry steam is 160 lbs. per square inch by gauge, and when it expands adiabatically in the nozzle and leaves it at a pressure of 2·4 lbs. per square inch. First use the formula given in the previous lecture, and then MacFarlane Gray's formula in this lecture, for finding the velocity of the steam. State the differences in your calculated results as a + or − percentage error from that found on page 244 in the previous lecture.

3. Give a short illustrated description of the impelling or torque action of steam on the moving blades of a continuous expansion, parallel-flow turbine Try to illustrate, not only the directions in which the steam passes the fixed and moving blades, but also show, by aid of graphic statics, the percentage resultant torque.

4. Give a neat freehand sectional elevation of Parsons' compound turbine, with a complete index to parts, show the several fittings and state their respective functions, as well as how they act in sequence.

5. Describe, with sketches, a Parsons' steam turbine. State why it is necessary to make the steam go in series through many elementary turbines.

6. Explain in outline the principle and arrangement of a Parsons' steam turbine. (I.C.E., *Feb.*, 1903) You may be guided in answering this question by Fig. 5, which shows the outside longitudinal view of the general arrangement of the turbine as applied to the driving of dynamos.

7. Sketch and describe the construction and action of the electric governing arrangements for a Parsons' turbine. What special functions does it perform, and how are these effected? Is it necessary to have a hand-lever connected to the rod coupling the governor with the emergency admission valve? If not, why not; if so, why?

8. Sketch and describe the construction and action of the mechanical governing arrangement for a Parsons' turbine. Which special functions does it perform when the engine is also fitted with an electric governor, and how are these severally effected? How may two centrifugal governors replace one such governor and an electric one? For which kind of applications are these two systems of governing specially adapted, and why?

9. Sketch freehand an outside view and longitudinal section of the Brush Parsons turbo-alternator. Place the necessary index letters at the most important parts, and describe concisely by aid of these letters the construction and action of the several parts of these machines. Give a short explanation of the course followed by the steam in passing from the admission valve to the exhaust pipe in this turbine. How is the leakage of air past the end glands prevented?

10. Sketch and describe the type of bearing used in small Parsons turbines. Enumerate the chief features of these "flexible" bearings.

11. State what you know regarding the relative floor spaces, heights, and volumes occupied by Parsons' turbines and reciprocating engines of the same power.

12. Give the necessary conditions for obtaining the best vacuum with steam turbines. State the relative amounts of cooling water which should be allowed during condensation at full-load steam consumption with a turbine, as compared with that of a reciprocating engine.

13. Illustrate and describe the principle and action of the vacuum augmentor. Explain clearly, by aid of a diagram like Fig. 12, how and why it is important to have a very good vacuum with steam turbines.

14. Plot on squared paper the data given on page 278 for the 500-kw. turbo-alternator.

15. Give a concise general description of the "Turbinia" and its turbines. Illustrate your answer by neatly-drawn sketches, and use index letters to represent the different parts.

16. Why is the efficiency higher with one propeller than with three propellers on each turbine shaft?

17. Give a freehand sketch of (*a*) an outside view, (*b*) a half outside view and half section, and (*c*) a section through the nozzles and blades of a Curtis turbine. Place the requisite index letters upon your sketches and give an explanatory index to the chief parts Show how the steam passes through the nozzles and blades, &c Also state how the various moving parts are made, lubricated, balanced, and governed.

18. Compare, by aid of sketches, the respective merits and demerits of the De Laval, Parsons, and Curtis turbines State for which kind of work each turbine is best adapted.

19. What advantages are claimed for the governing of the Curtis turbine over the Parsons turbine? What precautions are taken to prevent the possibility of the turbine being started before the oil begins to flow in the step bearing, and at what pressure is the oil supplied to this bearing? Give a ready estimate of the amount of oil required per minute for the bearings in this turbine

20. Enumerate the several advantages and the chief features of steam turbines.

21. Compare Prof. Rateau's empirical formula with the one given in connection with the De Laval and the other given in connection with the Parsons turbine Also check these formulæ by drawing an adiabatic curve to a scale of ·1 inch per lb. of pressure and per cubic foot of volume, when using dry admission steam of 215 lbs per square inch absolute and expanding down to ·93 lb. absolute. Now, ascertain the full area of your diagram of work. You may find the area of possible work done by any simple rule, or by use of a planimeter

LECTURE XVII.

LAND BOILERS, MECHANICAL STOKERS, FUEL ECONOMISERS.

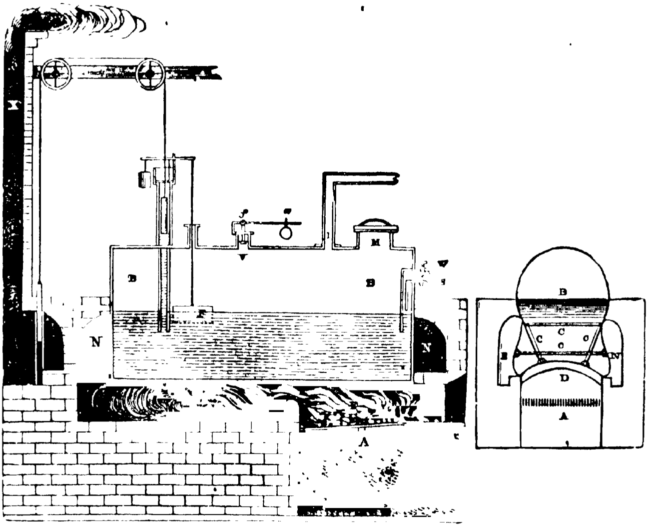
It is not our intention in this lecture to enter into the early history of Steam Boilers so fully as we treated the early history of the Steam Engine, since the former were not made with any regard to known mechanical principles, but merely of the form most convenient to the manufacturer.

Among the earliest types of boilers of any note, the old *spherical* boiler, and the *haystack* boiler of Smeaton, occupied foremost positions. The latter kind was largely used prior to the time of Watt, and consisted of a series of flanged cast-iron rings, which were fixed together by means of bolts passing through the flanges. The top of the boiler was hemispherical, and also of cast-iron, and the boiler was fired from beneath. Many of the engines of the Newcomen type, which were fitted up by Smeaton, were supplied with boilers of this kind.

Waggon Boiler.—The "*waggon*" boiler, so called from its resemblance in shape to a carrier's waggon, was introduced by Watt, and supplied by him along with most of his engines. It is illustrated in longitudinal and cross section in the following diagram taken from Prof. Rankine's book on *The Steam Engine.*

The top of this boiler was circular, and the sides and bottom concave, as shown by the cross section. The sides were sometimes made flat, but concave sides seem to have been more general. The fire-grate was situated underneath the boiler, and is shown at A. The flame and products of combustion passed from the fire-grate under the boiler to the end; here, they entered one of the lateral flues, N, and traversed along one side of the boiler, across the front end, and back along the other side to where they passed into the chimney. This arrangement of conveying the products of combustion round the boiler was known as the *wheel draught*, since the gases circulated from the back end right round the boiler. In the larger sizes of waggon boilers

made by Boulton & Watt, a flue was formed in the boiler itself. The heated products of combustion passed from the fire-grate underneath the boiler to the back end as before, but returned



B	for Boiler	M	for Man-hole.
A	,, Fire-grate.	V	,, Safety Valve.
N	,, Lateral flues.	F	,, Float.
S and W	,, Steam and Water cocks.	X	,, Chimney.

along this central flue to the front. Here the gases divided, and moved back to the chimney at the other end of the boiler through the lateral or side flues on both sides. This method of conducting the gases round the boiler was termed a *split draught*, since the gases divided at the front end of the boiler

The column of water in the vertical feed pipe formed the pressure gauge, and the damper was controlled by the rise and fall of this column. The water level was regulated by a float arrangement shown at F. The float rose and fell with the water level in the boiler. In falling it opened a valve in the vertical feed pipe, which admitted water, whilst in rising it closed this valve. This arrangement is one which is only suitable for the very low pressures (from 5 to 10 lbs. per square

inch) at which these boilers were worked, since it is evident that with high-pressure steam the vertical pipe would become inconveniently high.

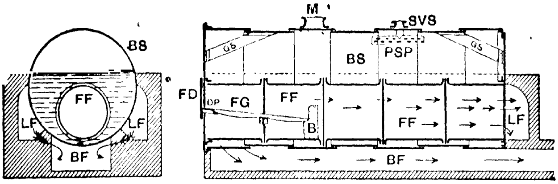
The waggon boiler was the style of boiler in most general use in this country for a long time, and it was not until about the year 1830 that it began to give place to forms more capable of resisting internal pressure. Its form rendered it suitable only for the very low pressures which Watt used, and even then it required to be stayed in the manner shown in the cross section in order to prevent deformation.

Egg-Ended Boiler.—When a higher pressure than about 15 lbs. per square inch became generally adopted, engineers were compelled to resort to cylindrical boilers. One of the first forms of cylindrical boilers tried was that known as the egg-ended boiler. This boiler was cylindrical with hemispherical ends. It was simple to construct, and required no staying whatever. The arrangements for firing and conducting the products of combustion round this form of boiler, were the same as those previously in use for waggon boilers, but there was the difficulty of obtaining sufficient heating surface. In order to obtain a moderate extent of heating surface, boilers of this kind required to be made very long in proportion to their diameter. It also had one serious defect, which was not found in the waggon boiler, viz., all sediment collected in the bottom of the boiler where the heat was greatest, and the plates were thus liable to get burned. In the waggon boiler the sediment collected in the two bottom corners, and these were not exposed to the most intense heat.

A modification of the egg-ended boiler, known as the "French" boiler (sometimes as the "elephant" boiler) was much used in France.

Cornish Boiler.—In order to increase the heating surface of egg-ended boilers, they were latterly constructed with a flue passing from end to end. The products of combustion from the fire-grate underneath the boiler, after moving to the back end, returned through the flue to the front end, and then passed back to the chimney along the lateral or side flues as in the improved waggon boiler. When, however, it was discovered that the radiation of heat from a boiler furnace amounted to 20 or 30 per cent. of the total heat of combustion, the furnaces were placed inside this flue, so as to impart to the water surrounding the flue, more of the heat radiated from the furnace. Boilers of this kind were introduced by Trevithick, and since they were first used to work the pumping engines at the Cornish mines, they

are known as *Cornish* boilers. The following diagram shows a longitudinal and a cross section through a Cornish boiler as now commonly constructed :—



CORNISH BOILER.

BS	represents the	Boiler shell.	BF	represents the	Bottom flue to chimney.
FF	,, ,,	Furnace flue.	GS	,, ,,	Gusset Stays.
FG	,, ,,	Fire-grate.	M	,, ,,	Manhole.
DP	,, ,,	Dead plate.	SVS	,, ,,	Safety valve seat.
B	,, ,,	Bridge.	PSP	,, ,,	Perforated Steam Pipe.
FD	,, ,,	Fire door.			
LF	,, ,,	Lateral or side flues.			

The products of combustion pass from the fire-grate through the flue to the back end of the boiler, where they divide and return to the front end along the two lateral or side flues. At the front, the products of combustion pass down to the bottom flue, and re-uniting move off to the chimney in contact with the bottom of the boiler. By this arrangement the gases are reduced in temperature before coming in contact with the bottom of the boiler, where all sediment collects, and there is, therefore, no danger of burning the plates on the under side of the boiler. Sometimes the gases are discharged direct from the furnace flue into the lower flue; but, unless in the case of very long boilers, where the gases may be considerably cooled before leaving the furnace flue, or where the water is very pure, this plan is objectionable, since the underneath plates on which sediment accumulates are liable to get burned.

The flues in the boiler shown are welded at the longitudinal joints, and the several rings are joined together by Adamson's flanged joints. The front end plate is attached by an outside angle iron ring, and the back end plate by an inside angle iron ring, and these end plates are stayed to the shell by gusset plates. The furnace bars are made in two lengths, and are supported at mid-length by a cross bearer. At the front end these bars rest upon the dead plate, and at the back end they

are supported by the fire-brick bridge. When air is admitted to the furnace flue at the bridge, a cast-iron stool usually supports the furnace bars and the fire-brick, and a suitable sliding door is fitted to the stool, the opening of which for the admission of air is controlled from the furnace mouth. As a rule, however, the whole of the bridge is constructed of fire-brick. The external flues are built of ordinary bricks, but are always lined in the inside with fire-brick.

Lancashire Boiler.—The Cornish boiler is only suitable for small powers, since the length of the fire-grate cannot be increased beyond that conveniently worked by firemen, which is usually from 5 to 7 feet, and a flue of large diameter is weak, unless made of very thick plates. Hence, when a large quantity of steam is required, the construction of the boiler is modified by having two flues of moderate size fitted into it instead of one. This, then, forms what is termed the Lancashire boiler. In every other respect it is exactly the same as the Cornish boiler.

In the Lancashire boiler the furnaces are usually fired alternately, so that while the one may be giving off smoke and unburnt hydrocarbon gases, the other shall be burning briskly, and with its greatest heating effect. By this arrangement, when the gases from the two furnaces mix in the external flues, the unburnt gases given off by the green fire are raised to the point of ignition by the greater heat of the gases from the brighter fire, and are thereby burnt by mixing with the excess of heated air which has passed through the other furnace.

Lancashire Boiler with Fox's Corrugated Furnaces and Flues (*see Coloured Plate*).—If the student will first of all compare each detail in the "index to parts" with the different views of this Lancashire boiler and then refer to these parts again when studying the following extracts from the specification, he should thoroughly understand its construction as well as the materials of which the more important parts are made :—

The Shell O S, to be 30 feet long by 9 feet in diameter, and $\frac{11}{16}$ inch thick, of Siemens mild steel. These plates to have a tensile strength of from 26 to 30 tons per square inch, with an elongation of 20 per cent. in a length of 10 inches. Each complete circumferential shell ring to be made in one plate. The seams to be butt-strapped inside and outside. All holes to be drilled and riveted together by means of six rows of rivets. These longitudinal seams to be clear of the brickwork. The circular seams to be double-riveted. The edges of all plates to be planed and caulked inside and outside, and the corners thinned by Penman's patent machine, so as to avoid fracture by local heating. The front and the back end-plates to be attached to the shell by their flanges.

Fox's Corrugated Flues F C F, to be two in number, 3 feet 8 inches outside diameter, and tapered down on the last ring to 3 feet $1\frac{3}{4}$ inches by $\frac{1}{2}$ inch thick, of Siemens mild steel boiler plates. These plates to have a tensile strength of from 24 to 28 tons per square inch, with an elongation of about 25 per cent. in a length of 10 inches.

End-Plates.—The boiler ends to be each of one plate, $\frac{3}{4}$ inch thick, of Siemens mild steel, and to have a tensile strength of from 26 to 30 tons per square inch, with an elongation of 20 per cent. in a length of 10 inches. The plates to be turned on their outer edge, and the holes for the flue ends to be bored out by special machines. These end-plates to be efficiently stayed by suitable gusset stay-plates G S, fastened by double angle steel to the shell of the boiler, but not brought down nearer to the furnace crowns than will give full freedom for expansion.

Manholes M_1, M_2.—One wrought-steel manhole door with double-riveted frame, and one of smaller size in the front end-plate, below the furnaces $F_1 F_2$. Both of these manhole doors to be riveted on and faced across the whole surface of their flanges.

The Stand Pipes to be made of wrought steel and riveted on the boiler where required. Their upper surfaces to be faced for connecting the various valves and cocks.

A Fusible Plug to be placed on the top of each furnace.

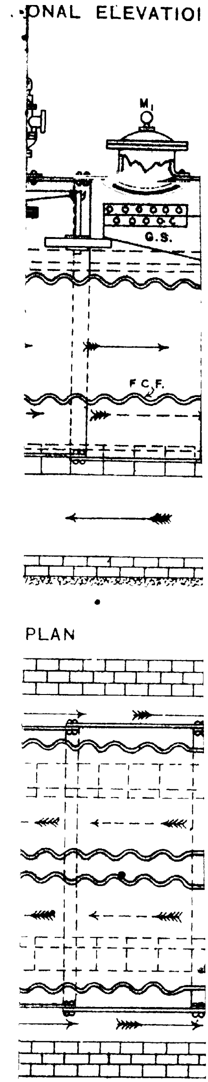
Furnace Fittings.—A set of five frames and doors to be fitted to the front end of the boiler, with brass beading round the outside of the furnace doors F D. Each door to be provided with an internal baffle plate and circular slide to regulate the admission of air for the prevention of smoke. Fire bars FB_1, are to be provided with bearers and cast-iron hearth-plates.

Dampers D_1, D_2 —Two cast-iron dampers and frames with pulleys, chains, and counterweights D W, complete.

Flue Doors C D.—Two flue doors and frames complete.

Blow-off Cock B O C.—One $2\frac{1}{2}$-inch gunmetal, asbestos-packed, blow-off cock, with solid bottom and packed gland, having a flange at each end. Also a suitable taper elbow, cast-steel pipe for attaching the cock to the stand-pipe.

Feed Valve F V.—A $2\frac{1}{2}$-inch check feed valve having the valve loose from the spindle, so as to act as a check or non-return valve, having a regulating hand-wheel and screw. A pipe for the effective distribution of the feed water to be fitted inside the boiler.



Iron Works, Glas

rts, $\frac{1}{2}''$ Ends, $\frac{3}{4}''$

Safety Valve.—One deadweight safety valve D W S V, and one balanced high- and low-level water safety valve B S V, to be fitted to their stand-pipes.

Stop Valve S V.—One 8-inch steam stop or junction valve to be fitted with a gunmetal valve and its seating, and to have a packed gland, with hand-wheel, &c.

An Anti-priming Steam Pipe S P, to be fixed inside the boiler and connected with its stand-pipe for the steam stop valve S V. This pipe to be perforated on its upper side.

Water Gauges G.—Two sets of ¾-inch gunmetal, asbestos-packed, water-gauge cocks with glass tubes and indiarubber washers.

Steam Gauge S G.—One 10-inch steam pressure gauge of best construction with siphon and cock.

Testing and Working Pressures.—Before leaving the works the boiler to be tested to a water pressure of 200 lbs. per square inch, and the every-day working steam pressure to be 120 lbs. per square inch.

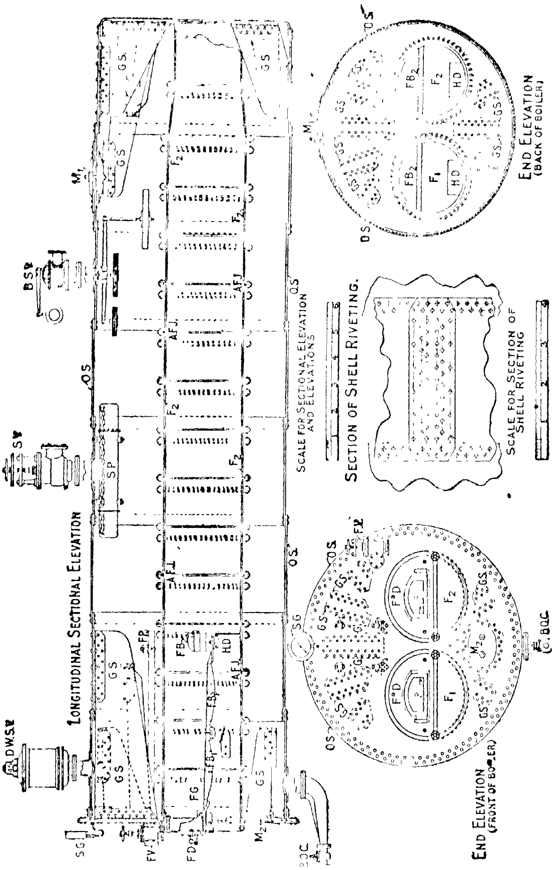
Specification of a High-Pressure Lancashire Steam Boiler. — *Shell.*—The outer shell O S, to be 30 feet long by 8 feet in diameter and 1 inch thick, made of mild selected steel boiler plates (Siemens process). Each ring in the circumference to be of one plate, thus removing all the longitudinal seams from the brickwork setting. The longitudinal seams to be butt-strapped inside and outside, and riveted together by means of six rows of rivets. The circular seams to be double-riveted The edges of all the plates to be planed and caulked both inside and outside. The corners of the plates to be thinned by means of Penman's patent machine, thus avoiding fracture by local heating.

End-Plates.—The boiler ends are each to be in one piece of mild selected steel boiler plate ⅞ of an inch thick. The front end-plate to be attached to the shell by means of a solid welded angle steel ring, 5½″ × 5½″ × ⅞″, fixed externally. The back end-plate to be attached by flanging. These plates to be turned up on their outer edge, and the holes for flue ends bored out by special machinery. The end-plates to be efficiently stayed by suitable gusset plates G S, fastened by double angle steel to the shell of the boiler. The stays shall not be brought down too near the furnace crowns, thus leaving a sufficient space for expansion.

The shell and end-plates to have a tensile strength of from 26 to 30 tons per square inch, with an elongation of 20 per cent. in a length of 10 inches.

Flues.—Two flues, F_1, F_2, to be made of ⅝-inch thick mild steel boiler plates (Siemens process), each 3 feet 3 inches diameter, and tapered on second last ring to 2 feet 9 inches diameter, with the last ring parallel. The first and last rings to be 1⅛ inch thick, and to have a tensile strength of from 24 to 28 tons per square inch, with an elongation of about 25 per cent. in a length of 10 inches. Each flue ring to be in one plate in circumference. The longitudinal seams to be solidly welded, and the transverse seams formed by flanging the ends of the plates, with a caulking strip between the flanges, as in Adamson's flanged joint A F J.

Manholes.—One M'Neil's patent manhole door M_1, with double-riveted frame, as recommended by the Board of Trade, to be fixed on top of boiler. One of smaller size M_2, to be placed on the front end-plate below the furnaces. Both manhole doors to be riveted on and faced across the whole surface of their flanges.



Lancashire Boiler (30 feet long by 8 feet diameter).
For a Working Pressure of 200 lbs. on the Square Inch.

Drillings.—Special machinery shall be used for drilling the whole of the plates after they are put into form. The pitch of rivet holes to be regulated, both longitudinally and transversely, by the dividing arrangement. Riveting, wherever practicable, to be performed throughout by hydraulic power.

Stand Pipes.—Wrought-steel stand pipes to be double-riveted on the boiler where required, and faced on their upper surfaces so as to connect the valves and cocks to the boiler.

Anti-Priming Pipe.—A cast-iron anti-priming steam pipe S P, to be perforated on the upper side, and to be fixed inside the boiler, and connected with the stand pipe for the steam stop-valve S V.

Fusible Plugs.—A fusible plug to be placed on the top of each furnace.

Furnace Fittings.—Fire frames and doors F D, to be fitted to the front end of the boiler, with brass beading round the furnaces. Each door to be provided with circular slide to regulate the admission of air. An internal baffle plate to be provided for the prevention of smoke. Cast-iron hearth plates, and fire-bars $F B_1$, of suitable length, with bearers, to be fitted in the furnaces and to rest on the cast-iron hinged doors H D, for the fire bridge $F B_2$.

Dampers.—Two cast-iron dampers and frames, with pulleys, chains, and weights complete.

Ashpit.—Ashpit frame and plate for front of boiler.

Flue Doors.—Two flue doors with frames.

Blow-off Cock.—One 2½-inch gunmetal, asbestos packed, blow-off cock B O C, with solid bottom and packing gland. A suitable taper elbow cast-steel pipe to be provided for attaching the cock to the stand pipe.

Feed-Valve.—A 2½-inch check feed-valve F V, with its valve kept loose from the spindle, so as to act as a check or non-return valve. The valve to be actuated by means of a hand wheel and screw. A feed-pipe F P, to be placed inside the boiler for effectual distribution of the feed water.

Safety-Valves.—One deadweight safety valve D W S V, and one high- and low-water safety-valve B S V, to be fitted to the boiler.

Steam Nozzle.—One 7-inch steam stop- or junction-valve and seating S V, with hand wheel for regulating the supply of steam from the boiler.

Water Gauge.—Two sets of ¾-inch gunmetal asbestos packed water glass gauge cocks G G, with india-rubber packing washers and glass tubes.

Steam Gauge.—One 10-inch steam pressure gauge S G, of best construction, with syphon and cock.

Testing.—The boiler to be tested with water pressure of 300 lbs. per square inch, and to be fit for a daily working steam pressure of 200 lbs. per square inch.

Water-Tube Boilers.—In water-tube boilers, sometimes called *tubulous* boilers, as distinguished from *tubular* boilers (of which the marine boiler is an example), the flame and hot gases from the furnace act directly on rows of parallel tubes of small diameter, which contain water and steam. These tubes are connected to a receiver from which a supply of steam is drawn. The principal advantages possessed by water-tube boilers over those of the ordinary cylindrical form of the same power are:—

(1.) Less weight.

(2.) Less space required, and their shape can be modified to suit the space available.

(3.) Much larger grate area.

(4.) Higher pressures can be employed without danger.

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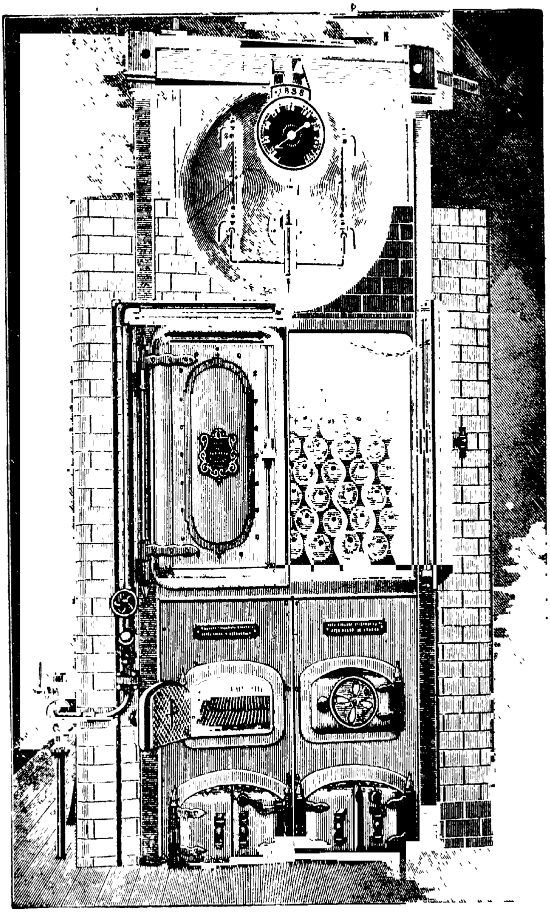
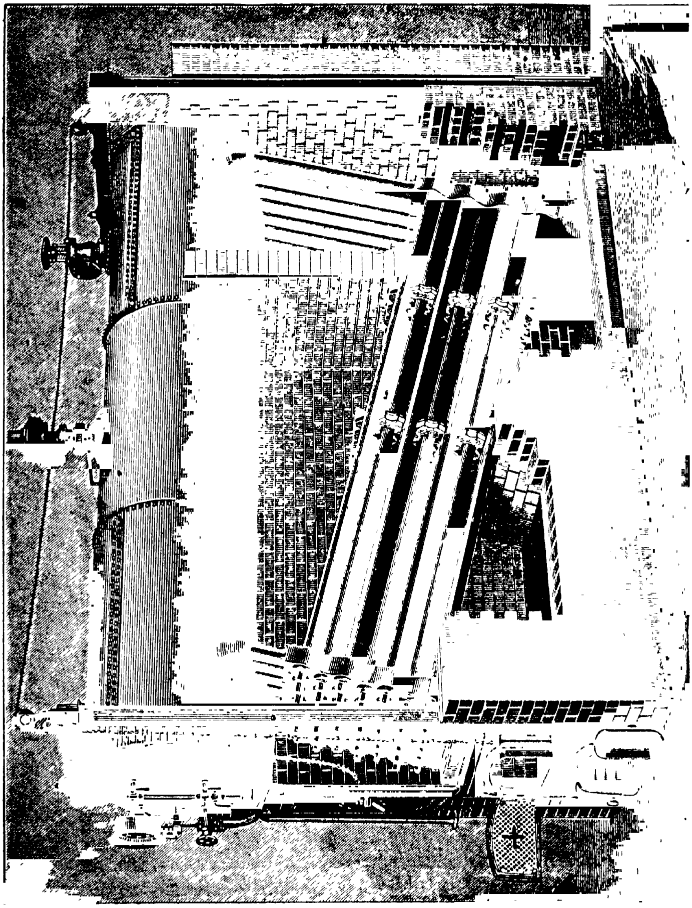


END VIEW OF THE BABCOCK & WILCOX BOILER.

(See Folding-Plate of this Boiler for the other views.)



SIDE VIEW OF THE BABCOCK & WILCOX BOILER

(5.) Larger and more subdivided heating surface, and, hence, more rapid steam raising.

(6.) If any part of the boiler should fail or be damaged in any manner, it may be replaced with very little trouble and in a very short time.

(7.) When used on board ship they can be removed in pieces without opening the deck.

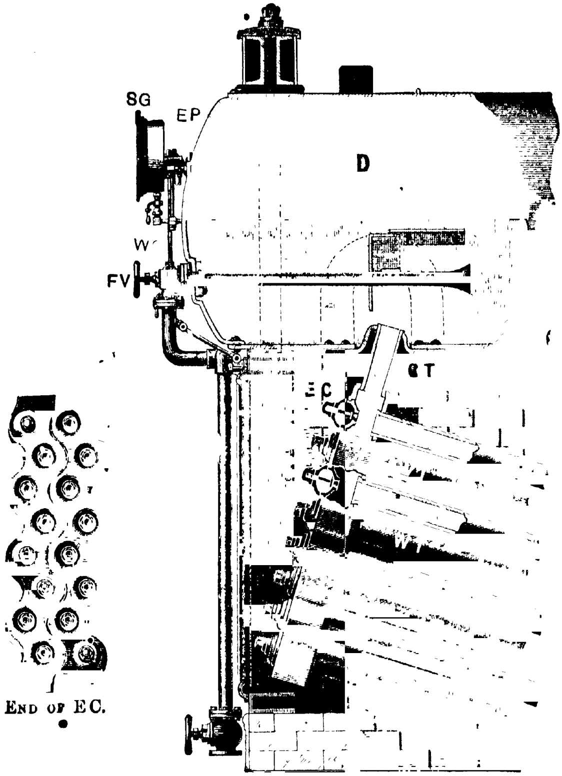
(8.) The several parts being comparatively small and light, they are easily transported and fitted up in places where it would be difficult to send a whole cylindrical boiler.

As an illustration of a water tube boiler, we have selected that manufactured by Babcock & Wilcox, Limited, London and Glasgow, and below we give report of a test made by Prof. Henry Robinson on one of five 140 H.P. B. & W. boilers at Regent Park Electric Lighting Station, St. Pancras Vestry, London,

Heating surface,	1,619 square feet
Grate area,	30 ,,
Ratio of heating to grate surface,	53·9 to 1
Kind of fuel—Nixon's navigation steam coal,	
Duration of test,	5 hours.
Average steam pressure,	173 5 lb.
Average temperature of feed-water,	48° Fah.
Pounds of coal fired,	2,352.
Pounds of refuse,	56.
Pounds of combustible,	2,296.
Per cent. of ashes,	2·38.
Coal consumed per sq. ft. of grate per hour,	15·6.
Total water evaporated,	22,925 lb.
Average evaporation per hour,	4,585 lb.
Maximum evaporation per hour,	5,160 lb.
Water evaporated per square foot of heating surface per hour,	2·83 lb.
Water evaporated per lb. of coal,	9·747 lb.
Water evaporated per lb. of coal, assuming feed-water at 212° Fah. and at atmospheric pressure,	11·906.
Temperature of boiler-room,	61° Fah.
Temperature of flue gases,	185° Fah.
Force of draught in inches of water,	0·391.

The boiler consists of several rows of water tubes, WT, placed in an inclined position (see next fig.). Each upright row of these tubes is connected together by suitable end connections E C, which are fitted to the water and steam drum D, at both ends, by upright connecting tubes C T. The water tubes W T, are not arranged vertically above each other, but are placed zig-zag, so as to break up the flame and products of combustion. Two

division plates consisting of iron, faced on the front sides with fire-bricks, fit between the tubes (see Plate, side view). These



The Babcock & Wilcox Water-Tube Boiler.

guide the flame and hot gases which rise from the fire-grate between the upper ends of, W T, and cause them to deflect down by the first fire-brick division, then to rise again behind the

second, and finally to pass across between the lower upright connecting tubes, and down again to the chimney tunnel thus . The water and connecting tubes (W T and C T) are 4 inches diameter, made of seamless steel. The steam and water drum, D, is made of wrought-steel plates, the longitudinal joints being double-riveted, and the circumferential joints single-riveted, while the end-plates, E P, are of steel and are dished out to the required curvature. Steam is taken from the drum, D, by the stop valve, which is fitted with an internal pipe, perforated with small holes on the upper side to prevent priming. Two glass water gauges, W G, and a steam pressure gauge, S G, and a combined feed and check valve, F V, are fitted to the front end of the steam drum, D, while a safety valve is fitted on the top centre, and a mandoor at the rear. Underneath the lower end of the water tubes, W T, there is a mud drum for collecting the sediment The mud drum is a wrought-steel box, fitted with sludge doors, and blow-out apparatus, by which it may be thoroughly cleaned out. The sediment should be blown out every twenty-four hours.

Babcock-Wilcox Steam Superheater (see Folding Plate and full-page figure). *General Arrangement.*—This consists of solid drawn steel tubes, 1½ inch diameter, bent into **U** shape and connected at both ends by expanded joints with wrought-steel cross-boxes or manifolds M_1, M_2 The upper box M_1, receives saturated steam from the drum D, by the inlet steam collectors S C_1; whilst the lower box M_2, collects the superheated steam after it has traversed the superheater tubes T, and delivers it by the outlet steam collectors S C_2, to the stop valve S V. As will be seen from the side elevation, hand holes H, are provided for getting into and cleaning M_1, M_2, as well as the tubes T.

Flooding.—The arrangement for enabling the tubes T, to be flooded with water consists of a pipe P_1, from the water space of the boiler drum D, to cocks C_1 and C_2. By opening the large upper cock C_1, and shutting the drain cock C_2, water passes from P_1 through P_2 into M_2, and thus fills the superheater tubes T. Any steam formed in these tubes during this operation is returned to the boiler drum by the collecting pipes S C_1.

Setting Superheater to Work.—When the full steam pressure has been attained and it is desired to connect the boiler to the main steam range of an installation, then close the boiler damper, shut cock C_1, and open drain cock C_2, which blows out all

INDEX TO

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BA [illegible]
WT [illegible]
H [illegible]
ST [illegible]
WG [illegible]
SG [illegible]
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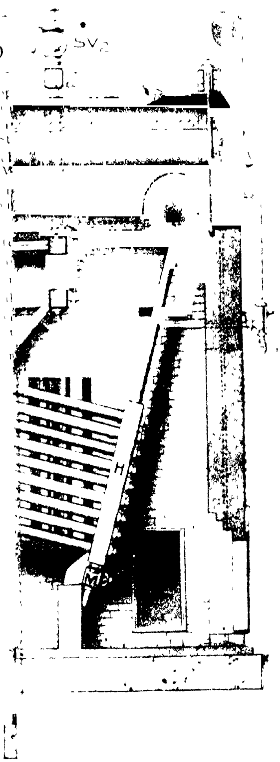
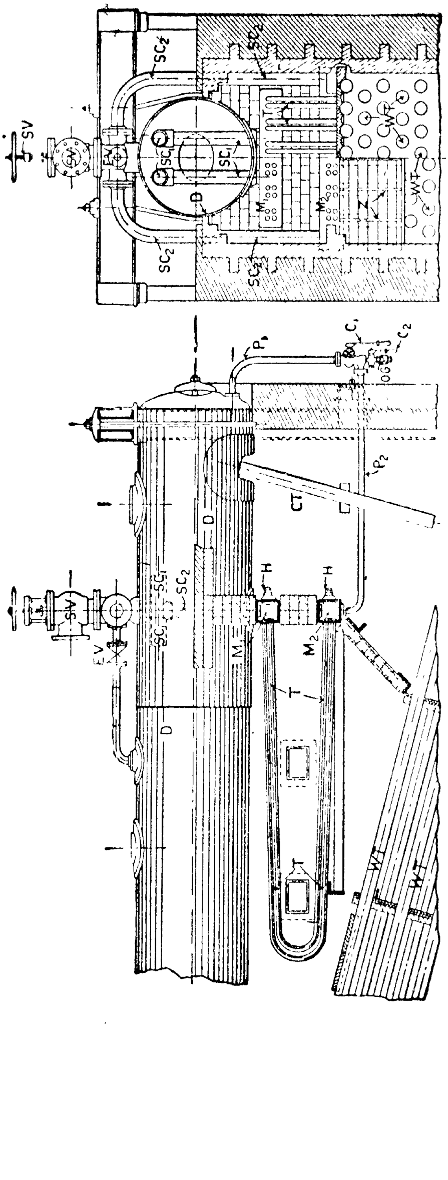


[illegible] CHAIN GRATE STOKER

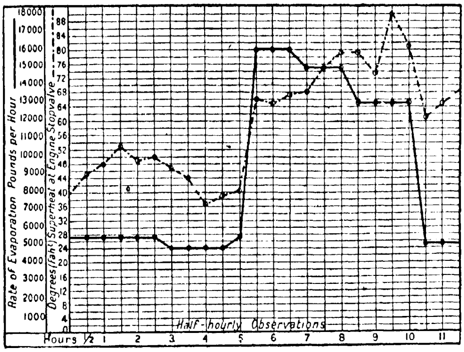


THE BABCOCK & WILCOX LAND BOILER
Fitted with their Steam Superheater

water from the superheater tubes T. When all the water is out, and the observation glass O G appears misty, shut C_2 and gently open the stop valve S V, as well as the damper. The saturated steam from drum D, is now conveyed by S C_1 to M_1, tubes T, M_2, and by the steam collectors S C_2, to the stop valve S V.

The equalising valve E V, should always be open when the stop valve S V, is shut, to prevent any syphoning-up of water when the superheater is flooded. Boilers fitted with a superheater should not take their feed-water by means of an injector.

Degrees of Superheat.—In this country, steam is ordinarily only superheated by 100° to 150° F., but, on the Continent, steam is frequently superheated up to 275° F., and this can be done in a standard Babcock & Wilcox superheater directly attached to the boiler, but with increased heating surface. With their independently-fired superheater, any desired higher degree of superheat may be obtained.



PLOTTED RESULTS OF EVAPORATION AND SUPERHEAT
With a Babcock & Wilcox Land Boiler.

Economy and Range of Superheating.—The accompanying curves were plotted by Mr. Arthur T. Cooper, Managing Engineer of the Reading Electric Supply Company, Limited, to show the results of his observations to ascertain the relation between the degrees of superheat obtainable under ordinary working conditions and the corresponding rate of evaporation of boiler water. It will be observed, that no special attempt was made to get a high degree of superheat, for the highest degree

attained was only 90° F. When the evaporation was as low as 5,000 lbs. per hour, the superheat was only 53° F.; whilst, with an evaporation of 16,000 lbs. per hour, the superheat was 68° F. The estimated working economy in coal consumption under these mild conditions amounted to about 12 per cent. over that without the use of a superheater.

Specification of the Babcock & Wilcox Boilers as Supplied to the Manchester Corporation Electricity Works.—There shall be two blocks, each comprising twelve boilers, each boiler having 3,580 sq. ft. of heating surface; the twelve boilers in each block being set in one continuous battery.

Sections.—Each boiler to be composed of 16 sections. Each section to be of 10 *seamless steel tubes*, 4″ diameter and 18′ long, connected at the ends by continuous wrought-steel staggered headers, or "uptakes" and "downtakes," into which the tubes are expanded. Each "header" to be provided with hand-holes placed opposite the end of each tube.

Headers.—To be of sufficient size to permit the cleaning, removal and renewal of a tube through the same. Each hand-hole to be provided with a cap fastened with a wrought-steel bolt, clamp, and cap nut.

Joints, Connections.—The several sections to be connected at each end with a mud-drum, by means of *seamless mild steel tubes* 4″ diameter and of suitable length, the joints being made with an expander. The two lower rows of tubes to be No 8 S.W.G. thick.

Drum, Manhole, and Cross-drum.—The steam- and water-drums to be 42″ diameter and 23′ 7″ long, made of mild steel plates $\frac{9}{16}$″ thick, with the longitudinal seams double riveted. A manhole to be provided, and fitted on to the shell, as well as nozzles for the safety valve, scum cock and attachment of cross-drum.

The steam- and water-drums to be connected at their rear ends by a riveted steel cross-drum 20″ diameter and 7′ 6″ long. The cross-drum to be provided with two wrought-steel standpipes for the attachment of the main steam stop valves, 6″ diameter, and one for the fixed soot cleaning valve which shall be 2″ diameter.

Lagging.—The exposed portions of the steam- and water-drums to be lagged to a thickness of 3″ with a plastic asbestos composition.

Mud-Drum and Blow-Off.—The mud-drum to be of wrought-steel, 6″square and 9′ 5″ long, with 4 hand-holes and a nozzle for blow-off pipe $2\frac{1}{2}$″ diameter.

Supports.—The front ends of the boilers to be supported from wrought-iron girders resting on brackets and secured to the columns of the building, and the rear ends of the boilers from wrought-iron girders, resting on wrought-iron columns with cast-iron bases. These columns to be properly secured so that the boilers shall be sustained entirely independent of the brickwork, and be free to expand or contract without affecting the same. The brickwork may be removed and replaced if required, without disturbing the boilers or connections.

Each boiler to be provided with the following fittings:—

Valves and Fittings.—Two 6″ diameter Turnbull's *sine quâ non* main steam stop valves.

Two 6″ diameter Hopkinson's isolating valves.

Two double 2″ diameter Turnbull's deadweight safety valves, set to blow at 210 lbs. per sq in. The valves to be enclosed, fitted with short bell-mouthed escape steam pipes, and suitable arrangement for easing them from the floor level.

Two $2\frac{1}{2}''$ Dewrance gunmetal combined feed and check valves with the screw on the outside.

Two $2\frac{1}{2}''$ Dewrance angle feed valves. Gunmetal bodies and outside screw with lengthened spindles.

One $2\frac{1}{2}''$ Dewrance globe feed valve for fixing between drums, with a gunmetal body and outside screw.

Two $2\frac{1}{2}''$ Dewrance scum cocks.

Two $2''$ gunmetal valves for the fixed soot cleaning.

One $18''$ dial Schaffer & Budenberg steam pressure gauge, graduated every 10 lbs. to 400 lbs., with syphon and control cock. The syphon to have a fitting for the attachment of the inspector's test gauge.

Two $\frac{5}{8}''$ Dewrance asbestos packed gunmetal water gauges fitted with automatic closing valves.

One fitting for the attachment of test pump

One fitting for the attachment of a Crosby indicator.

Note.—All valves to be clearly marked with the direction of opening and closing

Soot Cleaning—A suitable system of soot cleaning to be provided.

Scum.—A scum trough to be fixed in each steam- and water-drum and connected up to the $1\frac{1}{2}''$ scum cock on the drums.

Fronts.—The front of the boilers to be of ornamental pattern, arranged for the application of mechanical stokers, and with a large door for access to the ends of the tubes. All parts to be ample in strength and all joints to be fitted (*see end of this Specification for details of the Mechanical Stokers*).

Fixtures—The fixtures for each boiler to consist of a full set of flame bridge plates with bolts and special firebrick for lining the flame bridges; bridge wall girders and bars, binders and bolts, ash and cleaning doors for the access to the exterior of tubes and the flues for cleaning. Two doors for fitting in ash tunnel, two dampers with frames, and the requisite lintels for opening in walls, smoke chamber T's and anchor bolts for the front. The dampers to be suitably arranged for working from the front of the boiler The exposed portions of the ironwork to be given two coats of paint.

Gangways.—A gangway to be provided and fixed over the tops and in front of each block of boilers. The gangways in front of the boilers to be connected by a cross-over gangway, and all four gangways to be connected by two cross-over gangways running along the outside walls of the blocks of the boilers.

Two ladders to be provided for gaining access to the gangways from the boiler-house floor.

The gangways to be $16''$ wide and made of wrought iron with $\frac{1}{2}''$ diameter spills for the steps at $2''$ pitch. The gangways over the tops of the boilers to be supported by cast-iron stools resting on the boiler walls. The gangways in front of the boilers to be supported by brackets bolted to the building columns. The cross-over gangways at the ends of the boiler-house to be suitably supported from the ends of the blocks of the boilers.

Spares.—Four 2-cwt. iron coal barrows; two complete sets of firebars; six gaskets for manhole door in steam- and water drum; six gaskets for mud-hole door in mud-drum; six sets of fittings for hand-hole door in mud-drum; six complete sets of hand-hole fittings; seventy-two Jena gauge glasses with washers; one complete set of spanners in rack for gauge-glass fittings.

Tools.—Two wrenches fitting the hand-hole nuts; four tube scrapers with handles; eighteen tube brushes; four sets of fire tools, each set

consisting of three long and three short slices, three long and three short rakes, three long and three short prickers, with two dozen spare pricker blades; six tube stoppers; one 4″ expander, complete; two 5-cwt. Avery's movable platform weighing machines.

Testing.—The sections and mud-drum, also the steam- and water-drums, to be tested and made tight under a hydraulic pressure of 350 lbs. per square inch.

The boilers to be tested after erection to a hydraulic pressure of 350 lbs. per square inch.

Working Pressure.—The working steam pressure to be 200 lbs per sq. in.

Weight.—Approximate deadweight of each boiler—Gross 28½ tons, and net 27½ tons.

Brickwork.—The boiler brickwork above the foundation level to be provided by the makers. The boilers to be lined with 1½″ of firebrick throughout, and the top of the brickwork to be finished with bricks set on edge. A firebrick arch to be turned over the front part of the furnace. The end walls of each block of boilers to be of glazed brick, and the exposed brickwork in front of the boilers between the drums to be covered with ornamental cast-iron plates.

Mechanical Stokers.—Twelve pairs of Babcock & Wilcox patent chain grate stokers to be supplied for the afore-mentioned boilers (see Plate).

The grate to consist of an endless chain of short cast iron grate bars linked together, and actuated by passing over drums placed at the front and rear of furnace. The front drum to be revolved by means of a worm and worm wheel. Gearing to be provided to enable the whole stoker to be brought out clear of the front of the boiler. The frames to be fitted with wheels running on rails placed at the sides of the ashpit (*see left-hand figure of Folding Plate*).

Action of the Chain Grate.—The coal to be fed over the whole width of the grate, the thickness of the coal layer being regulated by the vertically sliding arrangement of fire-doors, which are so arranged as to allow (in case of need) of the stokers being fired by hand. The stoker, as illustrated, is able to burn small North Country coal without causing visible smoke at the top of a chimney 200 feet high.

The speed of chain grate to be capable of being varied from 5′ to 15′ per hour, to suit any demand for steam from the boiler.

The thickness of the fire and the speed of the chain can be adjusted to suit any class of coal or strength of draught.

The links to be made to travel in a body, with no relative movement between them, so that the coal may remain undisturbed until consumed.

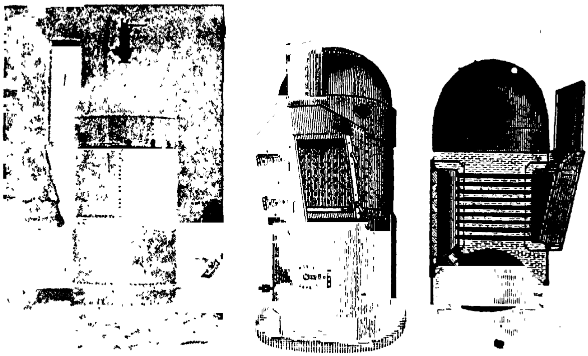
The links to be made to travel throughout the entire length of the furnace, and to deposit their hot ash at the rear into a clinker pit which is to be fitted with a dumping bottom. The links to be made to travel backwards underneath their drums, and to become practically cool by the time they reach the front end for another supply of coal.

Coal Hoppers.—The hoppers are to be arranged and placed at such a height from the ground that they can be filled from shoots coming down from the coal store, which is situated above the boilers, or by hand from the boiler-house floor. They are to be of sufficient size for convenient working by the latter means. The arrangement of the boiler fronts and the position of the water gauges, mountings, &c., to be such as will enable the coal shoots to be placed as indicated by the folding plate, without interfering with the working or obstructing the fireman's view of the pressure and water gauges.

Shafting for Driving Chain Grate Stokers.—The best quality of 4″ diameter mild steel bright shafting for working the chain grate to be supplied and placed above the boilers on plummer blocks, resting on cast-iron brackets, fixed to the boiler supports or to the boiler-house stanchions, as the case may be. (*The general arrangement of shafting is shown by the Folding Plate.*)

Vertical Boilers.—These are very useful for small isolated cases, such as supplying steam to winches, cranes, stone-crushers in quarries, and small river craft, or for donkey boilers on board steamers. They occupy a minimum of floor space, are self-contained, require no brickwork for flues or setting, and are very portable. They can be erected and set to work in a few hours; but, they are of necessity not so economical as the larger fixed boilers previously described, and they are liable to give off smoke when fired with bituminous and lower-grade coal.

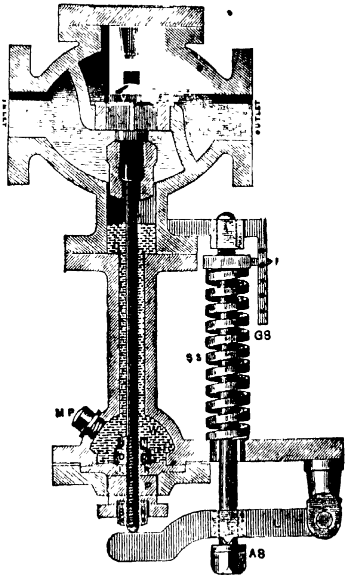
Cochran's Vertical Boiler.—As will be seen from the accompanying three small views of this type of vertical boiler, it consists of a circular fire-grate, hemispherical fire-box, vertical uptake, and horizontal flue tubes to the front smoke-box, upon which the funnel rests like a miniature marine boiler.



COCHRAN'S VERTICAL MULTITUBULAR BOILER.

The hemispherical fire-box and boiler roofs enable the makers to dispense with gusset or other stays. Since the horizontal flue tubes give a comparatively large heating surface and are freely accessible for cleaning both inside and outside, it is claimed that this boiler combines in a small space the advantages of the vertical form with the economy of the multitubular boiler.

As will be seen from the left-hand outside view, it is fitted with stop, safety, and feed-check valves, water and pressure gauges; water test, steam-jet and blow-down cocks, which are all tried on the boiler to a water pressure of 200 lbs. per square inch for 100 lbs. working steam pressure. When any one of these small boilers are likely to remain in an open, draughty, exposed position for a length of time, it is well worth the expense and trouble to have it carefully lagged with a non-conductor which will withstand rough usage and at the same time retain the heat. These boilers are seldom made to supply steam for more than 100 I.H.P., but they may be stacked and joined in series as a *"boiler battery,"* whereby each unit may be independently fired as required and put into or out of circuit for use or for repair.



AULD'S PATENT STEAM-REDUCING VALVE.

INDEX TO PARTS.

V	for Valve.	B N	for Boss nut.
W	,, Water.	M P	,, Mud plug.
J N	,, Jam nut.	A S	,, Adjusting screw
D	,, Washer.	S S	,, Spiral spring.
I R	,, India-rubber disc.	G S	,, Graduated scale.
P	,, Piston.	I	,, Index or pointer.

Auld's Steam-Reducing Valve. — Reducing valves are used for various purposes on land and marine steam plants where a lower pressure is required than in the boilers. The variety of designs for effecting this purpose is great, but the author has had practical experience with this selected example, and it will serve to explain their working to students.

The action of Auld's reducing valve is as follows:—The in-flowing steam is admitted between the valve V, and a piston P, which is covered by an india-rubber disc or diaphragm I R. The piston and valve are in equilibrium on the inlet or high-pressure side of the reducing valve A column

of water W (represented by dotted lines in the figure), is interposed between the steam and the india-rubber disc. This column of water is supplied by the condensation of the steam which takes place in the steam pipes. The desired pressure of steam on the outlet side of the reducing valve is obtained by compressing or relaxing the steel spiral spring S S (by the adjusting screw A S, which passes up through S S), until the index I, is opposite the figure on the graduated scale G S. The valve V, being thus opened, the steam flows through the valve opening to the low-pressure side of the reducing valve until the pressure of the steam on that side, pressing on the upper surface of the valve V, balances the force exerted by the spiral spring S S.

Should it be desired to obtain steam from the donkey boiler for engine-room machinery in the reverse way through the reducing valve, when there is no steam in the main boilers, then it is only necessary to compress the spring S S, by means of the adjusting screw A S, till the valve is opened.

When the reducing valve is used for supplying steam to weak steam chambers or pipes, a safety valve should be placed on the low-pressure side of valve.

These reducing valves are also serviceable for reducing and regulating automatically the pressure of water or air as well as of steam.

Prevention of Smoke.—At present the local authorities in most large manufacturing towns are seriously directing their attention to the prevention of smoke, not only from the furnaces of steam boilers, but also from ordinary house-fires. The number and the variety of the appliances which have been patented with this object in view are very great. We shall, however, only notice a few of those which have been brought specially under our notice of late.

In order to treat this matter fully and with the importance which it really deserves, we would require to devote to it a complete lecture on the analyses of different coals and their qualifications, the chemical actions which take place in the furnace and analyses of the products of combustion, together with the temperatures observed at different stages of the combustion and parts of the system such as in the furnace, smoke-box, tubes, flues, and chimney. Speaking generally, however, the emission of smoke from a furnace is a sign of imperfectly burned fuel. Consequently, if you wish to prevent the generation of smoke you must have as perfect combustion as possible of the coal and of the gases which naturally more or less arise from the burning of different classes of coal. To produce combustion of these smoky gases before they leave a furnace, they must be raised to a very high temperature, and be intimately mixed with the proper proportion of air. One of the simplest ways of doing this is to keep the back part of the boiler furnace always at a high degree of incandescence, and by so directing the smoky gases as they arise from the fresh green coal at the front end, that they shall pass down upon and over the incandescent surface at the back end. At the same time, the proper quantity of fresh air should be so admitted and directed from below the back furnace bars that it shall pass up through the incandescent coal and there become heated and mixed with the crude gases. In this way, you can induce a chemical combination which readily burns, leaving an almost smokeless flame to pass into the tubes or flues. The deposition of soot will thus be largely avoided in the flues and tubes of a boiler, and consequently the heat will be more efficiently and economically applied for the desired object of raising steam. There will be much less labour required in keeping

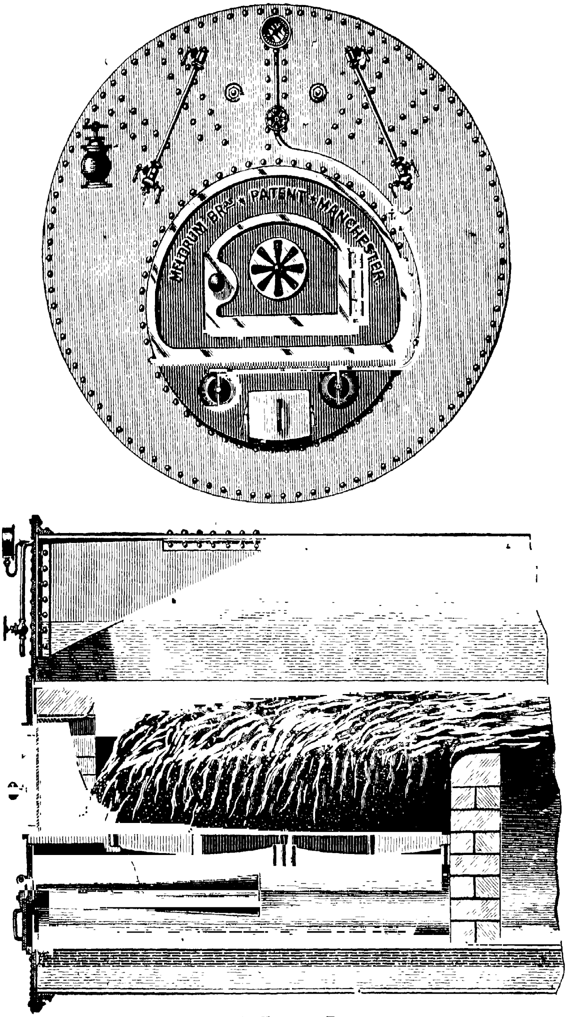
the heating surfaces clean, and the atmosphere will not become polluted to nearly the same extent.

A careful fireman, if provided with good coal, a properly proportioned furnace, and a boiler which does not require hard stoking in order to give the necessary amount of steam, can easily prevent the emission of smoke. The difference between the price of good steaming smokeless coal and dross, is a great inducement to steam users to employ the latter. With dross and the poorer cheaper qualities of smoky coal or with a boiler that requires frequent and hard stoking, even the most skilful fireman will be unable to prevent the making of smoke. In such cases, a good mechanical stoker or at least some form of forced or induced draught should be fitted to the boiler, and so arranged as to produce the aforesaid object of burning the gases before they leave the furnace flue (see *Index for examples of these appliances*).

With vertical boilers and house fires where it may be impossible to bring the smoky laden gases into contact with a highly heated incandescent surface, coke or gas-fires should be partially or wholly resorted to wherever convenient, or the smoke should be purified.

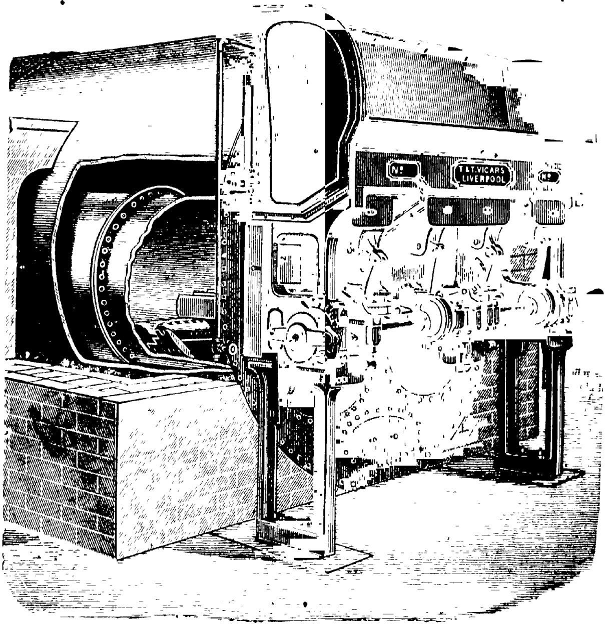
Meldrum's Forced Draught and Waste Fuel Furnace.—This furnace has been extensively applied for burning low classes of fuel, such as coal dust, coke dust, anthracite smudge, dross, &c. Since the price of fuel in many manufacturing industries constitutes such an important item in the cost of production, many persons are only too glad to avail themselves of any workable device which will enable them to increase their profits without diminishing the supply of steam from their boilers, and at the same time save them from coming within the range of the Authorities administering the Smoke Nuisance Act. As will be gathered by an examination of the accompanying end view and longitudinal section, Meldrum's furnace consists of an air-tight fitting cast-iron plate bolted to the mouth of the ashpit. This plate has a small air-tight door of square form, which may be removed when it is necessary to clean out the ashpit. There are also fixed to this door two conical blowers supplied with steam from a small pipe connected to the steam space of the boiler. The air-blast induced through the two tubes by the steam which issues from the two very small nozzles is regulated at will, by cocks placed in a handy position near the furnace door. This combined flow of steam and forced air up through the very narrow air slits between the very thin fire bars keep the latter comparatively cool, and thus prevents the adherence of clinkers. Besides the burning of inferior fuels, which is the chief object of Meldrum's furnace, it also enables boilers to be forced so as to supply any sudden or extra demand for steam, and it materially tends to the prevention of smoke. It is admirably suited (as we can testify from personal observation) for consuming the large and almost otherwise worthless quantities of coke riddlings produced in gas works. We have seen several of them at work in the Glasgow Corporation Gas Works, where the late manager, Mr. Wm. Foulis, M.Inst.C.E., had them applied, not only to Cornish and Lancashire boilers, but also to the water-tube boilers made by the Babcock & Wilcox Company.

Vicar's Mechanical Stoker.—As will be seen by an examination of the general view, sections, and index to parts, this stoker aims at feeding automatically into a boiler furnace small coal and slack at a regular rate from a divided hopper. The fuel is gradually pushed down from the hoppers on to the fire-bars by means of alternately reciprocating plungers actuated by eccentrics; and, then, by a slow reciprocating



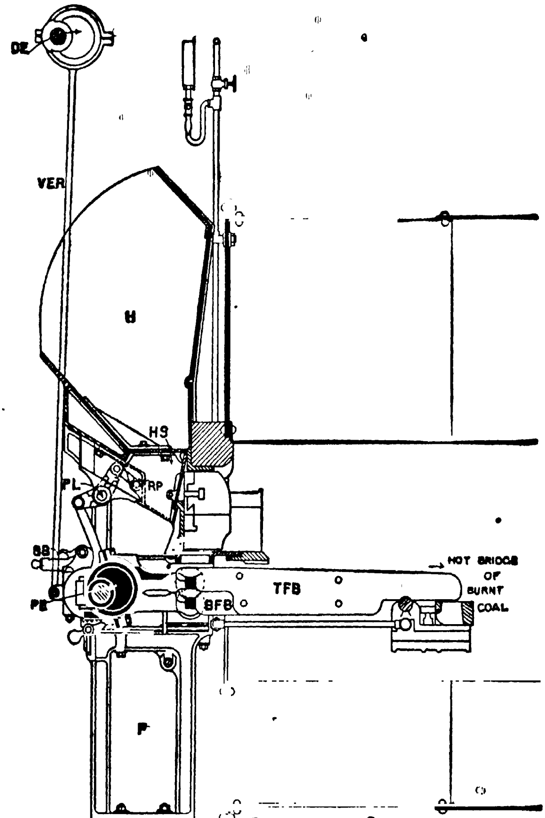
Meldrum's Forced Draught.

movement of the alternate sets of grate-bars, carried gradually forward, thus giving time for the fuel to become coked before it reaches the far end of the grate. Any unconsumed coal together with any clinkers and ash refuse are finally discharged over the end of the grate into the flues, where they are banked up and assume a highly incandescent mass closing up the far end of the ashpit. Over this white hot surface sweep the



VICAR'S MECHANICAL STOKER.

smoke laden gases, whereby their temperature is raised to such an extent that the fuel and the smoke are most thoroughly and effectually consumed. The supply of the fuel from the hoppers and the gradual movement of the same along the furnace is actuated by independent eccentrics turned by a central common shaft, driven from a small steam engine situated at the right-hand end of the shaft (not shown on the figures). The alternate fire-bars are first lowered and then drawn towards the front end of the furnace, and then all the fire-bars are moved simultaneously towards the back of the furnace This up and down travel of the bars may be adjusted from nothing to about 4 inches, by simply altering the leverage of the pall on the ratchet wheel which is turned by the driving eccentrics.



LONGITUDINAL SECTION.

VICAR'S MECHANICAL STOKER AND SELF-CLINKERING SMOKELESS FURNACE AS APPLIED TO A LANCASHIRE BOILER.

INDEX TO PARTS.

D E	represents	Driving eccentric.	P E	represents	Plunger eccentric.
V E R	"	Vertical eccentric rod.	S B	"	Stoking bar.
H	"	Hopper.	T F B	"	Top fire-bar.
H S	"	Hopper shoot.	B F B	"	Bottom fire-bar.
P L	"	Plunger lever.	F	"	Framing.
R P	"	Reciprocating plunger.			

The horizontal movements of the fire-bars are effected by cams on the central shaft, and they may be altered at pleasure from ½ to 2 turns of the shaft per minute. The whole framing and the gearing connected therewith are made quite independent of the boilers to which they may be attached.

The whole secret of smoke consumption by aid of these stokers lies in the fact, that the gases as they arise from the crude coal are raised to that degree of temperature and are supplied with that necessary quantity of air which will just induce them to become inflamable.

The following table of trials of Vicar's Mechanical Stokers as against hand firing with and without forced draught will prove interesting to the student:—

TRIALS OF THE VICAR'S MECHANICAL STOKERS AGAINST HAND FIRING WITH AND WITHOUT FORCED DRAUGHT.

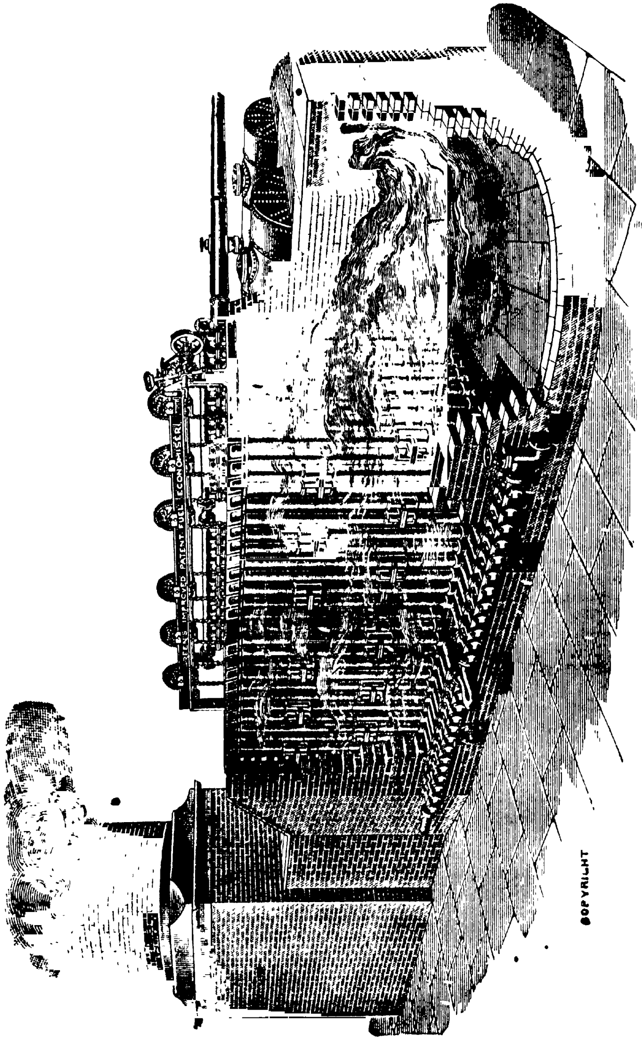
	Royal Paper Mills Wandsworth.		Vauxhall Water Works.	
	Vicar's.	Hand Firing Forced Draught.	Vicar's.	Hand Firing Ordinary Draught.
Number of Boilers,	1	1	4	4
Description of Boilers,	Lancashire	Lancashire	Cornish	Cornish
Dimensions of Boilers,	30′ × 7′ 5″	30′ × 7′ 5″	28′ × 6′	28′ × 6′
Dimensions of Boiler Flues, Diameter,	2′ 9″	2′ 9″	3′ 6″	3′ 6″
Area of Fire Grates (Total) Square Feet,	22	33	45·6	84
Number of Hours of Trial,	10	10	168	168
Designation of Coal,	Bituminous (Small)	Welsh (Small)	Bituminous (Slack)	Welsh
Coal Consumed (Total),	2 tons 13 cwts	3 tons 4 cwts	91 tons 16 cwts.	37 tons 10 cwts
Coal Consumed per Hour per Square Foot of Fire Grate,	27 lbs.	21 7 lbs	26 9 lbs.	6 lbs
Water Evaporated per Hour per Boiler,	5,330 lbs.	5,020 lbs.	2,525 lbs.	1,083 lbs.
Water Evaporated per lb. of Fuel,	8 98 lbs	7 lbs.	8·25	8·65
Price of Fuel per Ton,	13/-	15/-	13/-	16/6
Cost of Fuel per 10,000 lbs. of Water Evaporated,	6/5¼	9/4¾	7/4	8/6

Green's Fuel Economiser.—Messrs. E. Green & Son's Economiser consists of a series of cast-iron pipes, arranged in sections of various widths to suit existing circumstances, and placed vertically in the main flue through which the escaping gases from the boilers pass to the chimney. The pipes are 4 inches internal diameter and 9 feet long, and are connected together by top and bottom boxes and branch pipes. The exterior of these pipes is kept clean by sets of scrapers attached to chains actuated by the overhead gearing which is kept in motion by a small steam engine or water motor, all as shown by the accompanying general view. All joints are bored, turned, and pressed together with hydraulic machinery, and are now constructed to work at very high pressures. The feed-water is passed through the economiser on its way to the boilers, and thereby absorbs heat from the hot gases as they pass to the chimney. The saving in fuel varies, according to circumstances, from 15 to 25 per cent. as shown by the data in the accompanying tables.

Tests for the Value of Green's Economiser.—We are assured that the following tests were made for the purpose of showing the merits as a fuel saver of an "economiser." These tests were made at the different boiler plants, just as they were found in regular work, *no preparation having been made to obtain other than usual every-day results*, in favour of or against the economiser.

ECONOMISER WORKING OR NOT WORKING.		Economiser working December 15. *	Economiser not working December 16. *	Economiser working December 17. †	Economiser not working December 18. †
1. Duration of test,	hours	11·5	11·5	11·5	11·5
2. Weight of dry coal consumed,	lbs.	8,743	9,694	7,856	10,282
3. Percentage of ash and refuse,	per cent.	7·5	7·7	8·0	8·4
4. Weight of coal consumed per hour per square foot of grate surface,	lbs.	15·2	16·8	7·6	9·9
5. Weight of water evaporated,	,,	84,078	82,725	72,002	72,959
6. Horse-power on the basis of 30 lbs. of steam,	per H.P. hour,	247·0	243·5	210·3	213·6
7. Average boiler pressure (above atmosphere),	lbs.	68·2	67·2	58 0	57·4
8. Average temperature of feed-water entering economiser,	deg Fah.	84·2	...	88·	...
9. Average temperature of feed-water entering boilers,	,,	196·2	82·	225·2	85·2
10. Number degrees feed-water were heated by economiser,	,,	112·	..	137·2	...
11. Average temperature of flue gases entering economiser,	,,	435·	...	618	...
12. Average temperature of flue gases entering chimney,	,,	279·	452·	365·	645
13. Number degrees flue gases were cooled by economiser,	,,	156·	...	253·	...
14. Lbs. water evaporated per lb. of coal, as observed,	lbs.	9·617	8·533	9·165	7·096
15. Equivalent evaporation per lb. of coal from and at 212°,	,,	11·204	9·955	10·613	8·235
16. Percentage gained by using the economiser,	per cent.	12·5	...	28·9	...
17. Total area of heating surface in the plant,	square feet	3,126	...	2,804	..
18. Number of pipes in economiser,		160	...	160	..

* The steam in this test contained 1·3 per cent. of moisture. † The steam was superheated 50° Fahr., December 17th, and 52° December 18th.



GREEN'S IMPROVED PATENT FUEL ECONOMISER FOR STEAM BOILERS.

The Hopkinson-Ferranti Steam Stop Valve.—This comparatively new and novel form of stop valve was devised a few years ago by the well-known Electrical Engineer and Inventor, S. Z. de Ferranti, and it is made by J. Hopkinson & Co., Limited, Britannia Works, Huddersfield.*

The Principle of Action.—The principle upon which the valve acts is the same as that taken advantage of in the "Venturi Water Meter." And, therefore, both depend for their action upon the fundamental principle, as explained by "Bernouilli's Theorem,"* viz.:—When a fluid is flowing in a perfectly smooth pipe (of uniform or of varying cross-section), it possesses *kinetic energy* in virtue of its *motion*, in addition to the *potential energy* due to its *position* and *pressure energy* due to its head or *pressure*; and the *total energy* at any cross-section is the sum of these *three energies*.

Now, looking at the two figures opposite, we see from the longitudinal section, that if the steam comes from a boiler through a pipe of uniform section (at, say, 100 feet per second) towards the valve, then, when the valve is full open, the steam passes through a contracting nozzle leading to the throat. In doing so, its *pressure energy* is converted into *kinetic energy*, and the velocity of the fluid is increased in the ratio of the cross-area of the steam pipe to that of the throat, which is usually made 4 to 1. After the steam gets through the throat, it passes towards the engine along a diverging nozzle. In doing so, the *kinetic energy* is converted back into nearly the same *pressure energy* by recompression, and hence the steam is then *nearly* of the same pressure and velocity as before. That is, if the steam pipe on the outlet side of the valve is of the same diameter as the one from the boiler to the valve, and if the inner surfaces of the converging and diverging nozzles and of the throat are perfectly smooth

The student will observe that the weight of steam passing any cross-section must be the same. Consequently, the total heat energy in the steam must be the same at each cross-section of the nozzles and throat, *except* for any loss due to friction and eddying. Hence, if at any cross-section the pressure is less the velocity must be greater, and *vice versâ*.

The two most important things to be observed:—First, it is very important to have a perfectly smooth throat. This is accomplished by a special design of the eye-piece, which is truly guided whilst being drawn into a central position. Any want of smoothness in the throat causes most serious eddying of the steam, upsets the action, and results in a considerable drop of pressure through imperfect recompression.

Second, it is of great importance to have the outflow cone of suitable dimensions, so that there may be a minimum loss in kinetic energy, and hence of steam pressure.

The form adopted is the best compromise between what is theoretically right, and what can actually be constructed in practice, and this was determined by a series of careful experiments carried out by Mr. Ferranti.

At first, most practical engineers looked upon this stop valve as a perfect "paradox." But now that its principle is becoming better understood, whilst its action, construction, and many advantages are rapidly converting the most sceptical. Since the valve seats and discs are made of Hopkinson's "Platnam" metal, this stop valve is specially suitable for use with high-pressure superheated steam, even up to 750° F.

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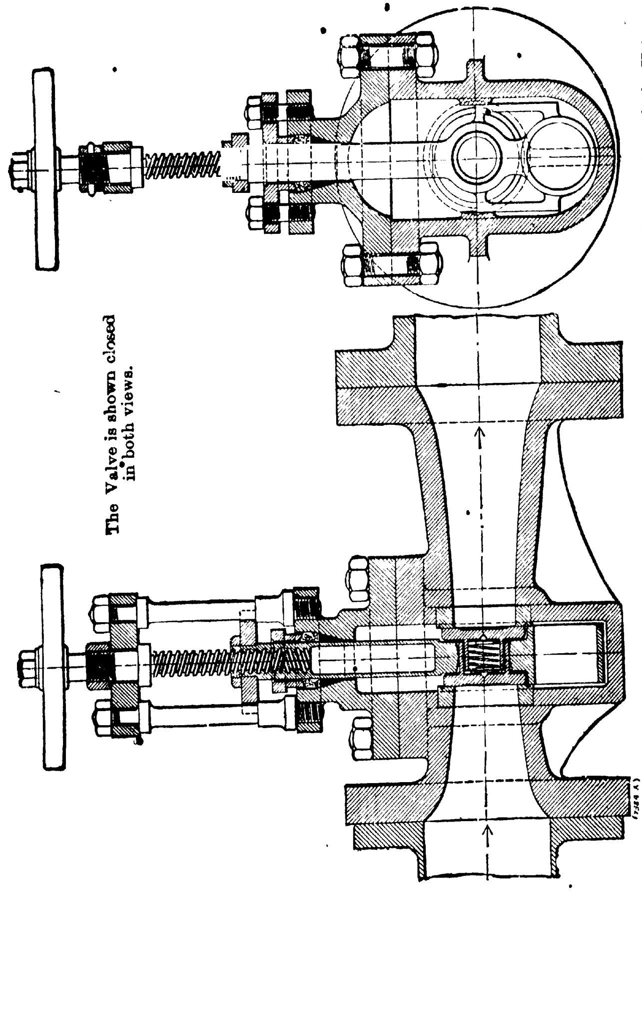
* See the Author's *Applied Mechanics and Mechanical Engineering*, Vol. IV. on *Hydraulics, &c.*, Lecture IV., Seventh or later Edition, where Bernouilli's Theorem is stated, illustrated, and proved mathematically.



Cross-Section through the Valve.

Longitudinal Section through the Valve.

THE HOPKINSON-FERRANTI PATENT STOP VALVE FOR STEAM BOILERS AND ENGINES.

LECTURE XVII.—QUESTIONS.

1. Sketch a section through a single tube or Cornish boiler, with a fire-grate inside the tube. Also describe, with a sketch, a safety valve, as applied to such a boiler.

2. Describe, with sketches, the construction of a Lancashire double-flued boiler. Show the position of the necessary fittings.

3. Sketch a section of a cylindrical land boiler, with two internal furnace tubes. Describe the mode of constructing the tubes so as to allow for expansion or contraction, and to prevent collapse. Show the manner in which the shell is strengthened by gusset stays.

4. Describe, with a sketch, an ordinary cylindrical land boiler with flat ends and internal flues. Enumerate the principal fittings and their uses.

5. Sketch the front view of a Lancashire boiler, showing all the necessary fittings. State the uses of the principal parts in giving an index of them. Also describe any apparatus for indicating the height of the water in land boilers with sketch and reference table.

6. Sketch and describe any form of breeches-flued boiler, and point out briefly its distinctive features. In what respect does this boiler possess an advantage over the Lancashire boiler?

7. Sketch and describe, with index of parts, any form of water tube boiler with which you are acquainted, and state what advantages this form of boiler has over ordinary cylindrical boilers. State also its disadvantages.

8. Describe fully by sketches a vertical cross tube boiler, and describe the several workshop processes by which such a boiler is constructed.

9. Describe the form of water tubes usually fitted to cylindrical land boilers, and show how they are fitted in.

10. Sketch and describe the best form of manhole for a boiler with which you are acquainted.

11. Sketch and describe two methods by which the internal tubes of a Lancashire boiler are strengthened against collapse. Name the advantages claimed for the plans you select.

12. Sketch a transverse section through a Lancashire double-flued boiler. Enumerate the principal external fittings, and their uses. Describe, with a sketch, the construction of the internal flues, stating the reasons for the particular arrangement shown.

13. Describe, with sketches, any boiler with its fittings and flues. Thus, for example, if you choose a Lancashire boiler, show the staying of the shells, the way in which the flues are helped to resist collapse, the brickwork seating, &c., Galloway or other flue tubes, how the feed water enters, and the steam is taken off, as well as the usual fittings. If the boiler has tubes, sketch and describe a boiler-tube ferrule, and say what are its advantages.

14. Describe, with sketches, a boiler of any kind with whose construction you are well acquainted. Describe the various kinds of joints adopted in its construction. Sketch the ends of the stays, or stay tubes if these are used. Sketch a stop-valve of a boiler, and show how the steam is taken away with the least chance of priming.

15. Describe a mechanical stoker and how it works. Under what circumstances is its use preferable to hand-firing?

16. Describe, with sketches, the construction of a vertical steam boiler, explaining carefully how additional heating surface to that of the firebox top and sides is usually obtained.

LECTURE XVIII.

MARINE BOILERS—CAUSES AND PREVENTIVE MEASURES FOR CORROSION OF BOILERS.

CONTENTS.—Rectangular, Oval, and Cylindrical Boilers—Single-Ended and Double-Ended Boilers—Boilers of S.S. *St Rognvald* with Specification—High-Pressure Boilers of S.S. *Arabian*—Boilers of S.S. *Inchdune*—Heating and Purifying the Feed-Water—Shanks' Small Vertical Marine Boiler for Steam Tugs—Corrosion of Marine Boilers, with Causes and Preventive Measures—Tables and Method of Testing Water for Corrosiveness—Questions.

Rectangular Boilers.—The old marine boilers were all made rectangular; and they continued to be made of that form so long as steam pressures below 35 or 40 lbs. per square inch were in use for marine engines. In modern practice, however, owing to the use of steam at a pressure greatly exceeding the above, boilers of this form are no longer manufactured, and few of them are in use except amongst the war-ships of the Royal Navy. As compared with the modern cylindrical boilers, working at the same pressure, they occupied less space in the ship for a given power, but they were heavier, owing to their form and the enormous number of vertical and horizontal stays required to support the flat sides; and being more difficult to construct they were therefore more expensive. They had, however, owing to the rectangular form, more steam space for the same amount of grate and heating surface. Rectangular boilers were constructed with any required number of furnaces, usually 3 or 4 for large powers, and the furnaces were arched on the top and bottom, and had flat sides. Boilers of this form were classified into *dry bottom* and *wet bottom* boilers. In dry bottom boilers, there was no water space underneath the furnace flues; there was in fact no bottom to them, simply recesses formed in the bottom of the boiler: whereas, in wet bottom boilers, the furnace flues were formed entirely inside the boiler, with a water space underneath. Rectangular boilers were as a rule tubular boilers, and the arrangement and position of the tubes were, except in a few special forms, similar to those in the modern cylindrical marine boiler—*i.e.*, the flame and products of combustion passed from the furnace flue into a combustion chamber at the back of the boiler, and returned through a series of tubes situated above the furnace flues into the smoke-box, and thence passed up the uptake to the chimney.

Oval Boilers.—To economise space and obtain some of the advantages of the old rectangular boiler, marine boilers are some

times made with flat sides and semi-circular at the top and bottom, and are known as *oval* boilers. The flat sides require to be bound together by transverse stays, which are made to pass between the rows of tubes. These boilers are simple to construct and work economically, and the thickness of the shell plates is less than would be required for a cylindrical boiler of same power and capacity, since the thickness is determined by the diameter of the semi-circular part at the top and bottom.

Cylindrical Boilers.—Modern steam boilers which have to resist very high pressures have their shells made cylindrical, since that is the only form for which staying is not necessary, and the flues are also made of this form. Cylindrical marine boilers are made either single- or double-ended—*i.e.*, the boiler is fired from one end or from both ends. The former contain from two to four furnaces, and the combustion chambers are variously arranged.

(1.) *In single-ended boilers*, when there are two furnaces there may be one combustion chamber common to both, or a separate combustion chamber for each furnace. The latter is the better arrangement, since, if any slight fault such as the leaking of a tube occurs in one combustion chamber, it may be repaired without the necessity of drawing both fires and blowing off steam.

When the boiler has three furnaces, there are almost invariably three separate combustion chambers, since, no other equal division can be arrived at.

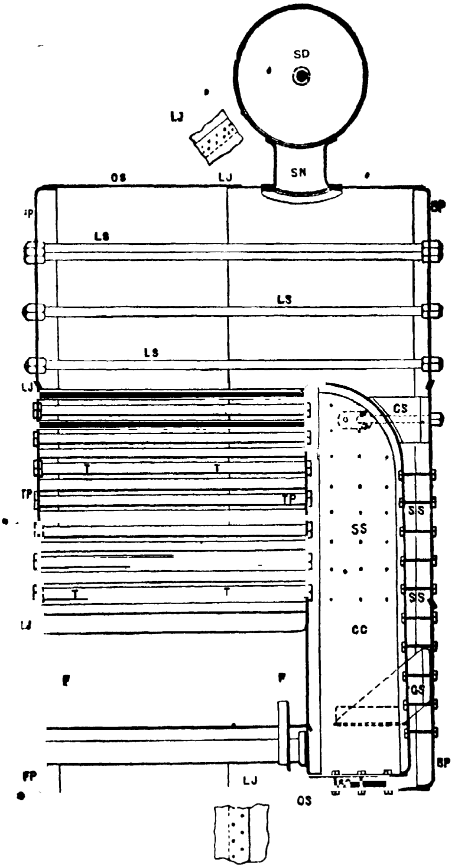
If there are four furnaces in the boiler there are usually either two or three separate combustion chambers. When two combustion chambers are fitted, each pair of side furnaces communicates with the same combustion chamber; and if three combustion chambers are fitted, the two central furnaces have a common combustion chamber, and the side furnaces have separate combustion chambers.

(2.) *In double-ended boilers*, the combustion chambers are arranged independently of the number of furnaces.

A very common and efficient plan is to have opposite furnaces connected to the same combustion chamber. Another arrangement gives one common combustion chamber for each end set of furnaces; while in the third, which is the heaviest and most expensive method of all, each furnace has a separate combustion chamber.

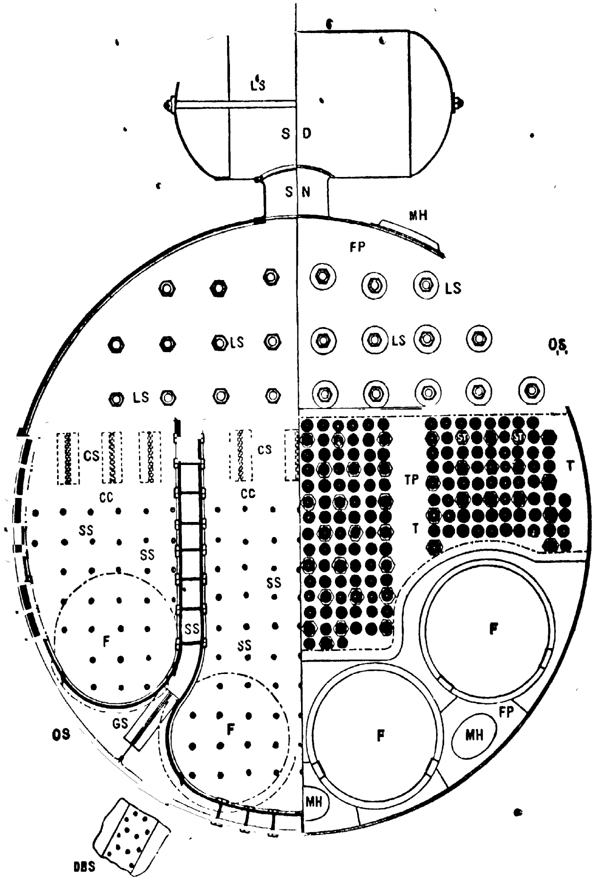
The arrangements of combustion chambers given above, are those usually met with in actual practice, although other plans may sometimes be adopted.

Boilers of S.S. "St. Rognvald."—The following illustrations



LONGITUDINAL SECTION – SCALE ¼ INCH = 1 FOOT.

O S	for Outside Shell.	T P	for Tube Plates (front and back),
F	,, Furnaces.	B P	,, Back-end Plate.
C C	,, Combustion Chambers.	L J	,, Lap Joint, in circumferential seams.
T	,, Tubes.		
F P	,, Front-end Plate.		



HALF-END VIEW AND CROSS SECTION—SCALE $\frac{1}{4}$ INCH = 1 FOOT.

DBS for { Double-riveted Butt Straps, with double straps for longitudinal seams.
SD „ Steam Dome.
SN „ „ Neck.
MH for Manhole and Mudhole Openings.
LS „ Longitudinal Stays.
ST „ Stay Tubes.
GS „ Gusset Stays.
CS „ Crown Stays.
SS „ Screwed Stays.

represent the boilers of the S.S. *St. Rognvald*, constructed by Messrs. Hall, Russell & Co., of Aberdeen.

SPECIFICATION OF MAIN BOILERS S.S. "ST. ROGNVALD."

General.—To be two in number, cylindrical, multitubular, and fired from one end with four furnaces in each. To be made of mild steel (Siemens' process), with the exception of the tubes.

Shells.—Each boiler to be 15 feet extreme diameter, and 10 feet 5 inches long. The plates to be $\frac{7}{8}$ inch thick, in two lengths fore and aft. The circumferential seams to be lap-jointed and double-riveted with rivets $1\frac{1}{8}$ inches diameter, and $5\frac{1}{4}$ inches pitch; the longitudinal seams to be made with double butt-straps $12\frac{1}{4}$ inches broad, $\frac{3}{4}$ inch thick, and double-riveted with rivets $1\frac{1}{8}$ inches diameter and 5 inches pitch.

The end plates to be $\frac{27}{32}$ inch thick, flanged all round, and double-riveted to shell. The whole of the rivet holes to be drilled 1 inch diameter before the plates are bent, and after they are bent they are to be fitted together and the holes drilled out in place to fit the rivets.

The edges of plates to be planed all round, and the seams of shell to be carefully caulked inside and outside. A baffle plate to be fitted to the fronts of each boiler *above* tubes.

Furnaces.—To be four in number for each boiler, 3 feet 4 inches outside diameter, and 7 feet long, plates to be $\frac{9}{16}$ inch thick, and the top plate to be in one piece and jointed to the bottom plate by double butt-straps $\frac{5}{8}$ inch thick and single riveted. The whole of edges of the plates and butt-straps to be planed and caulked outside and inside.

Combustion Chambers.—The two centre furnaces to have one combustion chamber common to both. The two side furnaces to have each a separate combustion chamber. The back and side plates to be $\frac{5}{8}$ inch thick, and stayed with screwed stays $1\frac{5}{8}$ inches diameter at bottom of thread, and pitched $8\frac{3}{4}$ inches apart; made of mild steel, and fitted with nuts at both ends. The tops of these chambers to be curved to the arc of a circle of 26 inches radius.

Tubes.—To be of iron, lap-welded, 249 in number (in each boiler), and $3\frac{1}{4}$ inches external diameter, and No. 9 B.W.G in thickness, swelled at front end to $3\frac{3}{8}$ inches diameter. The stay tubes to be 75 in number (in each boiler) $3\frac{1}{4}$ inches external diameter, and $\frac{5}{16}$ inch thick. These tubes to be screwed into back tube plate, and fitted with nuts on combustion chamber side, and to be secured into the front tube plate with nuts on each side.

Stays.—The longitudinal stays in steam space to be $2\frac{5}{8}$ inches diameter at bottom of thread, made of mild steel and pitched 16 inches apart. Washers $7\frac{1}{2}$ inches diameter, $\frac{1}{2}$ inch thick, to be fitted to each of the stays at both ends of boiler (outside). The end plates of the boiler to be tapped, and stays screwed in, to a good fit, and afterwards caulked. The whole of the staying to be sufficient for a working pressure of 90 lbs. per square inch.

Steam Domes.—One on each boiler 3 ft. 6 inches diameter, and 7 ft. long, with plates $\frac{1}{2}$ inch thick. The longitudinal seams to be lap-jointed and double riveted; the circumferential seams to be lap-jointed and single riveted.

The end plates to be $\frac{3}{4}$ inches thick, and dished to a circle 24 inches radius, and fitted with one steel stay in the centre, $2\frac{3}{4}$ inches diameter at bottom of thread. The domes to be connected to the boilers by strong neck pieces 16 inches diameter inside, and $1\frac{3}{4}$ inches long, and double riveted to shells of dome and boiler.

Manholes.—To be cut in the shells of the boilers where required, and to be fitted with wrought-iron doors, studs, bridges, &c., and compensating rings of flat plate, or angle-iron fitted round them, and double riveted to shell.

Testing.—The complete boilers and steam domes to be tested with water pressure to 180 lbs. per square inch, before leaving the works, without any leakage or signs of weakness.

Boiler Fixings.—The boilers to rest on very strong wrought-iron seats riveted to the ship's floors, with double angle irons on the floors. The seats to be well stayed in a fore and aft direction. The upper part of boilers to be securely fastened to the ship's beams, in such a way as to allow for the boiler expanding without opposition from the stays.

Lagging.—After the boilers have been fixed in the vessel and tested to 90 lbs. steam pressure, their upper parts and the steam domes are to be covered with an approved non-conducting composition, to extend round as far as the centre of the wing furnaces, and then to be sheathed with sheet lead or galvanised iron, bound with strong iron hoops.

Boiler Mountings.—Each boiler to have two spring loaded safety valves $4\frac{11}{16}$ inches diameter, with easing gear led to engine-room platform, one steam stop valve 7 inches diameter, one valve for steam to winches and cranes $2\frac{1}{4}$ inches diameter, one valve for steam to whistle, one surface blow-off valve, one bottom blow-off valve, one main feed check valve, one donkey feed check valve, one salinometer cock, two sets of asbestos packed gauge cocks; also an efficient means of circulating the water in boilers while steam is being got up. The whole of the above valve chests to be of cast-brass, with the exception of the safety valve and stop-valve chests.

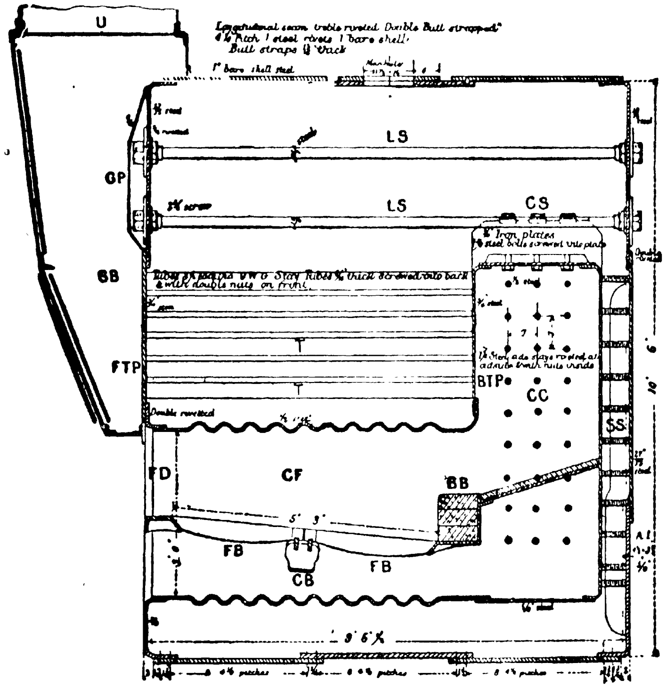
Furnace Mountings.—The furnace fronts, doors, and centre bar-bearer, to be made of wrought-iron, and the dead plates and bars of cast-iron. Furnace fronts below dead plate to be fitted with damper doors, with a rack to keep them open to the desired amount. A wrought-iron door to be fitted to lower part of bridge bearer in each furnace, so that ashes or coal thrown over the bridges may be removed.

Uptake and Funnel.—The uptake to be formed of $\frac{1}{4}$ inch and $\frac{5}{16}$ inch wrought-iron plates, with an air space of 2 inches between them. The smoke-box doors to have shield plates both outside and inside, and very strong hinges with brass pins riveted in. The funnel to be formed of $\frac{1}{4}$ inch and $\frac{5}{16}$ inch plates, 43 feet high from firebars, and 6 feet 6 inches diameter, with all necessary hoops and shackles for stays, &c.

Boilers of S.S. "Arabian."—As an illustration of boilers which work at a very high pressure, we have selected that shown in the following diagrams, which is the boiler of the S.S. *Arabian* (the engines of which were described in Lecture XXIII., Vol. I.), and was constructed by Messrs. Rankin & Blackmore, Greenock. The engines are of the triple-expansion type, and the boiler pressure is 150 lbs. per square inch.

This boiler is single-ended, and is 10 feet 6 inches diameter, and 9 feet $5\frac{1}{2}$ inches long. It has two furnaces, 3 feet diameter (minimum), constructed of Fox's corrugated steel plate $\frac{1}{2}$ inch

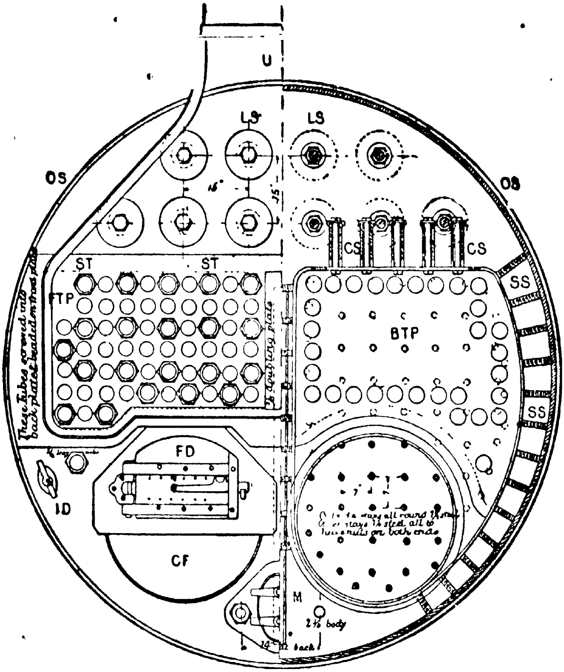
thick, and each furnace has a separate combustion chamber. The shell plates are of steel 1 inch thick, and the shell is constructed in three rings, which are united together circumferentially by double-riveted lap joints. These rings are jointed



• LONGITUDINAL SECTION—BOILER OF S.S. "ARABIAN."

longitudinally by treble-riveted butt joints, provided with double straps or cover plates, $\frac{13}{16}$ inch thick. The rivets used are of steel, 1 inch diameter, and, in the case of the longitudinal joints, they are placed at $4\frac{1}{8}$ inches pitch. The end plates are made in three pieces, and are fixed together by double-riveted lap joints, and flanged to meet the shell and the corrugated furnace flues. The furnace flues are flanged at the back ends, and riveted to the combustion chambers. The combustion

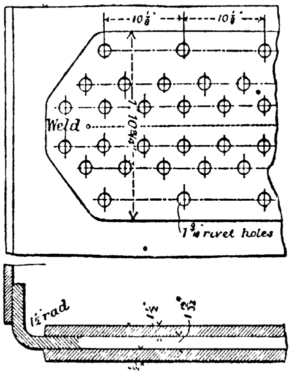
chambers are flat on the top, and deformation is prevented by girder-plate crown stays, C S, fitted with three $1\frac{3}{8}$ inch steel bolts for each stay. The screwed stays, S S, at the backs of the



BOILER OF S.S. "ARABIAN."

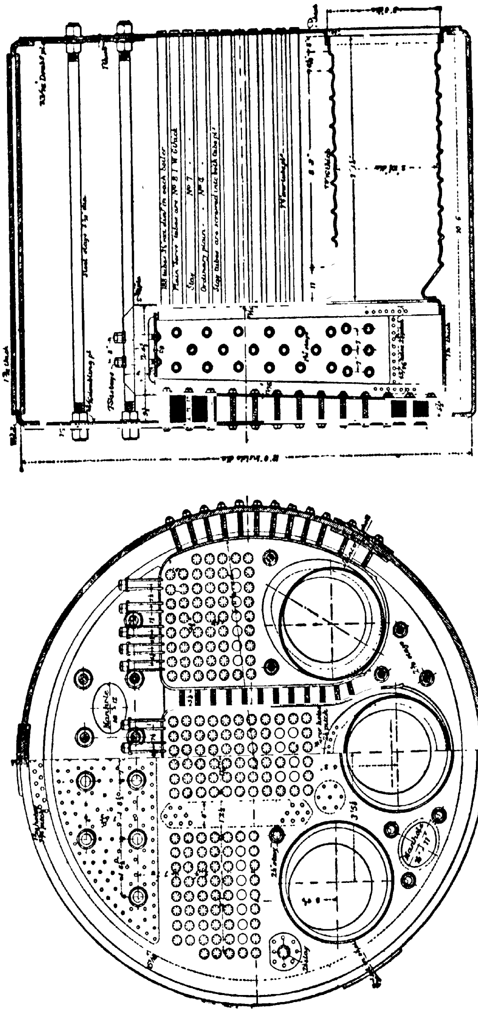
O S	for Outside Shell.	B B	for Fire Brick Bridge.
C F	,, Corrugated Furnace Flues.	M	,, Manhole.
C C	,, Combustion Chambers.	I D	,, Inspection Door.
T	,, Tubes.	L S	,, Longitudinal Stays.
S T	,, Stay Tubes.	C S	,, Crown Stays.
F T P	,, Front Tube Plate.	S S	,, Screwed Stays.
B T P	,, Back Tube Plate.	S B	,, Smoke-Box.
F D	,, Furnace Door.	G P	,, Guard Plates for L S nuts
F B	,, Furnace Bars.	U	,, Uptake to Funnel.
C B	,, Cross Bearer for furnace bars.		

combustion chambers, are of steel 1¼ inch diameter, with the exception of those forming the outside row, which are 1½ inch diameter. All these stays have nuts on both ends. The flat sides of the combustion chambers are stayed by 1¼-inch steel stays with nuts on the inner ends only, the outside ends being riveted over. The boiler contains 122 tubes T, 3¼ inches diameter, of which number 40 are stay tubes S T, whilst six others are screwed into the back tube plate B T P, and beaded over on the front ends. The stay tubes S T, are screwed into the back tube plate B T P, and have double nuts on the front ends. The stay tubes are $\frac{5}{16}$ inch thick, and the ordinary tubes, No. 9, W.G. thick, or ·148 inch. The upper part of the boiler, and also the lower part below the furnaces, are stayed by steel longitudinal stays L S, 2⅜ inches in diameter at the body, with 2¾-inch screws. A stay tube running through to the front plate below the smoke-box, is also provided on each side of the boiler, as shown in the end elevation, to support that part of the combustion chamber into which it enters. The manhole on the top of the boiler is fitted with a ring 6 inches broad and 1 inch thick, to compensate for the loss of strength of the plate in which the hole is cut.



DETAIL VIEWS, SHOWING METHOD OF WELDING THE BOILER PLATES FOR S.S. "INCHDUNE."

Boilers of S.S. "Inchdune."—These boilers are of the ordinary, single ended, Scotch type, two in number 13 feet in diameter



END VIEW, CROSS AND LONGITUDINAL SECTIONS OF THE BOILERS FOR THE S.S. "INCHDUNE" AND S.S. "INCHMARLO."
(*See Drawings and Description of the Engines.*)

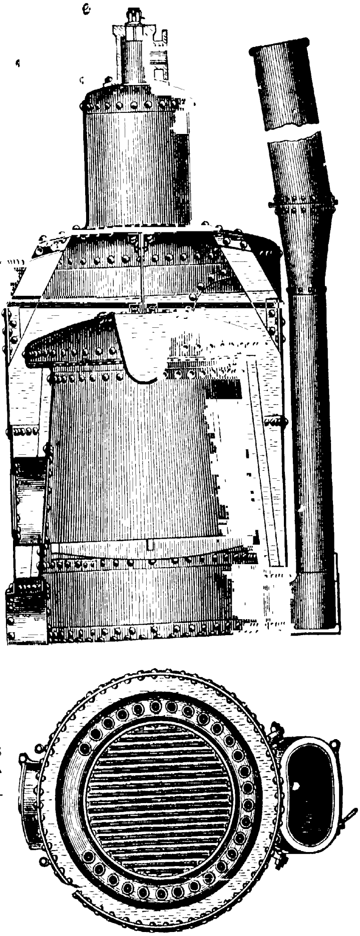
by 10 feet in length. They are constructed to carry the very high steam pressure of 267 lbs. per square inch. This high pressure necessitates the use of very strong stays and a shell plate $1\frac{19}{32}$ inch thick. The shell is made of three plates, butt-jointed, with inside and outside covering plates. The shell is flanged at each end, and, to render this possible, the plates were welded for a short distance, as shown by the detailed views. Since the loss of heat by radiation may be very great, extra attention was paid to the lagging of the boilers and the steam pipes. The starboard and port boilers are worked with Ellis and Eaves' system of induced draught. The induced draught is effected by two fans placed in the uptake to the funnel. These fans remove the spent gases from the boiler tubes, and cause a partial vacuum into which the air for combustion enters by passing through the furnaces. The efficiency of the boiler is still further increased by the use of Serve tubes and arch-ribbed furnaces.

Heating and Purifying the Feed-Water.—In all high-pressure boilers special attention should be given to the treatment of the feed-water before it enters the boilers. The feed-water should be of as high a temperature and as free from impurities as is economically possible. It is also important that the internal surface of the boiler should be protected against oxidisation. For this reason, the air held in suspension in the feed-water should be got rid of. To effect this, the feed is pumped from the hot well through a pair of filters to a contact heater, as made by G. & J. Weir, of Glasgow, where its temperature is raised to about 223° Fah., and where any air it may hold is eliminated. From this contact heater it is passed by means of a pair of Weir's feed-pumps through a surface heater to the boilers. The temperature of the feed-water when in the surface heater should be about 400° Fah., or nearly that of the steam in the boiler.

Small Vertical Marine Boiler for Launches and Tugs.—In Lecture XXI., Vol. I., we illustrated and described a small pair of non-condensing marine engines, by Messrs Shanks & Son, of Arbroath, for use in steam launches, yachts, and small screw-tugs. It is of importance in such cases to minimise the longitudinal space occupied by the boiler as much as possible, and with this object in view, the above makers have adapted their vertical multitubular boiler described in the last lecture.

The accompanying illustration will still further explain the internal construction of this form of boiler, as a sectional plan is shown in addition to the vertical section.

Great heating surface, easy repair, non-liability to prime, tubes easily brushed out, and the fact, that any sediment which may collect in the central fire-box tube can be quickly removed by simply opening a blow-off cock fitted on the outside shell to which is attached an external syphon pipe, are several of the advantages claimed for this arrangement Further, no fire-brick deflectors or internal linings are used, neither is the shell weakened by cutting it away to admit the tubes, nor bolted together by flanges, while the centre of gravity of the whole boiler is kept as low as possible to insure stability in the small vessels for which it is adapted. The whole boiler is made of Siemens-Martin steel.



SHANKS' VERTICAL MARINE BOILER FOR LAUNCHES, YACHTS, AND SMALL TUGS.

Corrosion of Boilers—Causes and Preventive Measures.*—The principal causes of corrosion in marine boilers are:—(1) Sea water, (2) animal and vegetable oils, (3) air, (4) galvanic action.

(1) Sea Water.—Sea water varies in composition to a very considerable extent, but the following analysis may be taken as giving about the mean average quantities of the various constituents:—

ANALYSIS OF SEA WATER, WITH PERCENTAGES OF THE VARIOUS AMOUNTS OF SOLID MATTER.

Substance	Grains per Gallon.	Lbs. per Ton	Percentage of Total Solid Matter.	Percentage per Gallon of Sea Water.
Carbonate of lime ($CaCO_3$),	9·20	·285	·4	·013
Sulphate of lime ($CaSO_4$), .	112·45	3·478	4·7	·16
Sulphate of magnesium ($MgSO_4$), . . .	160·26	4·957	6·7	·23
Chloride of magnesium ($MgCl_2$), . . .	261·91	8·101	10·9	·374
Chloride of sodium (NaCl),	1,870 86	57·871	77·3	2·673
Total, . . .	2,414·68	74·692	100·0	3·440

N.B.—Seeing that there are 70,000 grains in one gallon, or 10 lbs., of water, each of the above quantities are supposed to be free from any H_2O or water.

Under certain conditions, sea water in immediate contact with heated copper, brass, iron, or steel surfaces becomes acid, by the conversion of the chloride of magnesium into hydrochloric acid and magnesia. The hydrochloric acid dissolves a certain quantity of iron from the boiler surfaces, forming chloride of iron; as soon as the chloride of iron is formed it is decomposed by the magnesia already liberated, precipitating oxide of iron and reforming chloride of magnesium. The oxide of iron deposited is ferrous oxide and is black, and remains so unless air is allowed to get into the boiler, when it becomes ferric oxide and changes in colour from black to brown or red.

If corrosion is to be prevented sea water must be kept out of the boilers, and this can only be attained by making and keeping the condensers tight.

Rendering Sea Water Non-Corrosive.—To render the sea water which contains the above substances non-corrosive, about 8 lbs. of quicklime or 45 lbs. of soda crystals per ton would be required. Such large quantities are, of course, prohibitive when there is any considerable leakage of sea water with the boiler feed

* The author is indebted to Babcock & Wilcox, Limited, for the kind permission to abstract a quantity of this useful information from their excellent treatise on their "Water-Tube Marine Boilers." In reckoning the percentage of total solid matter and percentage per gallon of sea water, he did not think it was worth while to take the exact weight of a gallon of sea water as 71,820 grains as against 70,000 grains of pure water.

The use of lime has the further disadvantage, that out of the 8 lbs. mentioned above, $5\frac{3}{4}$ lbs. would be deposited in the form of scale. Sea water itself has sufficient sulphate of lime to produce $3\frac{1}{2}$ lbs. of scale for every ton, so that, if the proper proportion of lime be added, altogether the quantity of scale which will be deposited is $9\frac{1}{4}$ lbs. Although this scale, when evenly coated over the boiler surfaces, may protect them from the corrosive action of the chloride of magnesium, it is at the best only an expensive and unsatisfactory remedy, as it increases the consumption of fuel, and may damage the boiler by overheating when accumulated.

Evaporators.—Evaporators are essential for making-up the loss of water due to leakage at glands and joints, but they must be blown down before the brine becomes too concentrated; otherwise the chloride of magnesium will be decomposed and give off hydrochloric acid, which will pass over into the boilers with the distilled water, and thus render the fresh make-up feed-water acid. The action of the acid formed in this way differs from that formed in the boiler by the decomposition of sea water, inasmuch as it does not immediately afterwards become destroyed by re-uniting with the magnesia, but is carried in with the fresh make-up feed, and is held in solution throughout the entire volume of boiler water. It is thus in a position to attack all parts not protected by scale or otherwise. This is a point which demands careful attention, and is not sufficiently recognised by marine engineers. On no account should the brine be allowed to get above $\frac{3}{32}$. *When this point is reached, the evaporator should be blown down and refilled with fresh sea water.*

Dock and Tidal Waters.—A very objectionable practice is to fill and start the boilers with dock, river, or tidal waters. Dock water is a tidal water, and always contains a certain percentage of sea water, with other variable impurities.

Soda and Lime Correctives.—As will be seen from the foregoing remarks, the use of either soda or lime should receive careful consideration. It is difficult to give any definite instructions as to the extreme limit to which it would be wise to go in the case of introducing either soda or lime, since so much depends on the circumstances of each case, such as the density of the water, rate of working, and the time which must elapse before the boilers are next cleaned.

Experience, however, points to the use of lime as being the more generally satisfactory on a voyage. Soda may be used in cases where the boiler water is acid through the density of the brine in the evaporator being excessive, and where no vegetable oil has been allowed to enter the boilers; or, when entering port at the end of the voyage.

A small amount of salt water is bound to get into the boilers, even under the most favourable conditions, through priming of the evaporator, or due to a slight leakage in the condenser tubes. It is an excellent plan to continually use a small quantity of milk of lime to neutralise this sea water. One lb. of lime dissolved in fresh feed-water per 1,000 I.H.P. per day in the following manner may suffice, and the lime thus used should be the ordinary unslaked lime of commerce. It should be finely powdered and kept in a dry place. Then, the "milk of lime" is made by mixing about 1 lb. of this lime in a gallon of fresh pure water. It should be strained through wire gauze before use, in order to get rid of any lumps or solid impurities.

Anti-corrosive for New Boilers.—When starting with new boilers on a voyage for the first time, 5 lbs. of lime should be put into the boilers through a manhole for every 1,000 I.H.P. Then, 2 lbs. of lime per day for every 1,000 I.H.P. should be passed through the hot well, as milk of

lime, for about six days. For the remainder of the voyage about 1 lb. per 1,000 I.H.P. per day. At the end of the voyage the boilers should be examined to see if they have a thin coating of lime scale on their internal surfaces. If this is not the case, and the water shows a black or red colour, the use of lime should be continued.

Daily Tests.—The boiler water should be tested daily by the graduated testing bottle or salinometer, and, if found to contain a larger amount than about 100 grains of chlorine per gallon, the use of lime should be increased.

If the boiler water at any time be found acid, a solution of carbonate of soda should be added to the feed at the rate of a bucket of soda solution per hour until the water just turns red litmus paper blue, after which daily additions of soda or lime should suffice to keep the water in a safe alkaline state.

Carbonate of soda is effective in changing sulphate of lime into sulphate of soda, which is soluble and therefore harmless Carbonate of lime, which is also formed, may be easily blown or washed out.

In all cases on entering port, soda crystals dissolved in fresh pure water should be added to the feed, as this will tend to soften the scale and render the boilers more easily cleaned.

The use of soda *at sea* in boilers into which vegetable oil has been allowed to enter is sometimes attended by trouble, on account of the soapy scum which forms on the surface of the water being carried over into the H.P. cylinder by priming. In such cases lime alone should be used.

(2) **Animal and Vegetable Oils.**—Another cause of corrosion in boilers is the introduction of animal or vegetable oil with the feed-water. By using such oils as lubricants in the steam cylinders, the exhaust steam carries it over to the condensers. Such oil, containing fatty acids, will decompose and cause pitting wherever the sludgy deposit can find a resting place in the boilers.

Use of High Flash-Point Lubricating Oils.—Only a minimum quantity of the highest grade of hydrocarbon oil (such as the best "valvoline") should be used in the steam cylinders. In lubricating piston and valve rods, this same precaution should be observed Apart from the evil effects of acidity, the hydrocarbon deposited upon the heating surfaces of the boiler is most harmful. A thin film of this deposit forms a non-conductor, which prevents the heat passing through to the water, and causes the heating surfaces to burn, blister, and crack.

Filtering.—The feed-water should be purified on its way to the boiler by passing it through an efficient cleansing filter.

Graphite is sometimes used in place of hydrocarbon oils as a cylinder lubricant. In fact, graphite is generally superior to oil, and especially so when the steam pressure is as high as, say, 275 lbs. per square inch, which corresponds to a temperature of 400° F.; but, in the case of highly superheated steam at, say, 500° to 600° F., then it is found to clog the piston-rings.

Many marine engines are run without a particle of internal cylinder lubrication, except, what may be used in swabbing the piston-rods with pure hydrocarbon oil, under the care of careful competent engineers.

(3) **Air with Feed-Water.**—Air has been a well-recognised cause of corrosion for many years. Many instances of rapid corrosion have been proved to have been caused by the feed-pumps sucking air from the hot well, and the feed being delivered to the boiler at a level considerably below the water line.

Small bubbles of air expelled from the water on boiling, attach themselves tenaciously to the heating surfaces. The oxygen in the air at once

begins war on the iron or steel, and forms iron rust, making a thin crust or excrescence which, when washed away by the circulation or dislodged by expansion and contraction, leaves beneath a small hole or pit. "Pitting," once started, progresses rapidly, as the indentations form ideal resting places for the bubbles of air, and at the same time present increased surfaces to be attacked.

Fresh water at 32° F. absorbs 4·9 per cent. of its own bulk of oxygen; at 50° F. 3·8 per cent., and at 68° F. 3·1 per cent.; whilst salt water absorbs more air than fresh water.

To prevent the introduction of air into the boiler, the hot-well water should be pumped to a filter tank situated 8 to 10 feet above the feed-pump suction valves. By so doing, a large amount of air rises and is liberated from the surface of the water, and a head of water at the suction valves of the pump is assured. Care should be taken to keep the pump glands tight, and to efficiently entrap any free air in the air vessels.

(4) **Galvanic Action.**—Formerly, nearly all corrosion in boilers was attributed to this cause, and zinc slabs were suspended wherever possible within the water space. The position of zinc relative to that of iron in the scale of electro-positive metals, causes the zinc to be attacked in preference to the iron or steel of the boiler whenever galvanic action takes place. Zinc is, however, only attacked when the boiler water contains salt, because it is then electro-positive to the iron inside the solution. This action is merely another evidence of the presence of sea water in the boiler feed, and the fitting of zinc plates is only an expedient to minimise the action of the objectionable sea water. If the sea water be prevented from entering the boiler, the zinc will not readily act, and there will be little necessity for using it in large quantities, thus lessening a very expensive item in the working of marine boilers.

Prevention by Zinc Slabs.—To afford efficient protection by the use of zinc, there must be perfect metallic contact between the zinc slabs and the iron or steel shell of the boiler. The bolting of zinc slabs inside the drums of tubular boilers, and near the entrance of the feed-water, is recommended as of positive benefit. In fact, so long as the zinc slabs continue to oxidise and disintegrate within a boiler, it shows that they are confining to themselves an amount of harmful action which would otherwise act upon the iron or steel of the boiler.

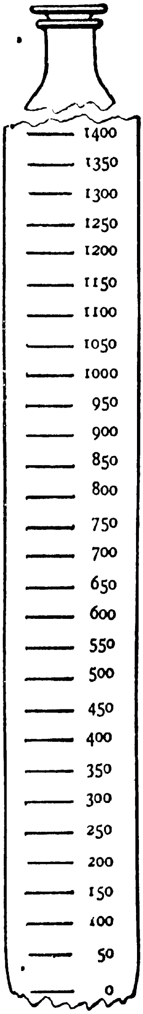
Method of Testing Water for Corrosiveness.—The first thing is to see that the colour of the sediment of the water, as shown in the gauge glass, is neither *black* nor *red*. The only colour admissible is slightly dirty grey or straw colour. So long as the sediment of the water is *red* or *black*, corrosion is going on, and it must immediately be neutralised by the intelligent use of lime or soda, with frequent scumming or blowing off of some of the boiler water. The "make-up" feed-water is provided for by the evaporators.

The ordinary salinometer is an instrument for determining the total quantity of solid matter in the boiler water. The apparatus here described gives a convenient and correct method of ascertaining the exact number of grains of chlorine in the water to be tested. It contains one graduated bottle, one bottle of silver solution containing 4·738 grains of silver nitrate to 1,000 grains of distilled water, and one bottle of chromate indicator, which is a 10 per cent. solution of pure neutral potassium chromate.

It must be clearly understood, however, that this apparatus merely determines the amount of chlorine which the boiler water contains per gallon. The solid matter per gallon corresponding to the chlorine is given in the following table:—

TABLE SHOWING PROPORTION OF CHLORINE AND TOTAL SOLID MATTER IN VARIOUS DENSITIES OF BOILER WATER.

Ord. Salinometer. Ozs. per Gall. 5 ozs. = $\frac{1}{32}$.	Chlorine. Grains per Gall. At 60° F.	Total Solids. Grains per Gall. At 180° F.	Admiralty Hydrometer Scale.
	1,400	2,310	
5 – Density of Sea Water	1,350 1,330 1,300	2,227 2,187 2,145	10°
	1,250	2,064	
	1,200	1,980	
	1,150	1,897	
	1,100	1,815	
4	1,050	1,734	8°
	1,000	1,650	
	950	1,567	
	900	1,485	
	850	1,402	
3	800	1,320	6°
	750	1,237	
	700	1,155	
	650	1,072	
	600	990	
	550	907	
2	500	825	4°
	450	742	
	400	660	
	350	577	
	300	495	
1	250	414	2°
	200	330	
	150	247	
	100	165	
	50	82	
	0	0	0°



To make the Test.—Fill the graduated bottle to the zero mark with the water to be tested; add one drop of the chromate indicator and shake the bottle; then slowly add the silver solution; keep shaking the bottle. On nearing the full amount of silver solution required, the water will turn red for a moment, and then back to yellow again when shaken. The moment it turns red and *remains red*, stop adding the silver. The reading on the graduated bottle at the level of the liquid will then show the amount of chlorine in grains per gallon. For example, if a permanent red colour is shown when the level is midway between 150 and 200, there are 175 grains of chlorine per gallon.

The principle of the process depends upon the fact, that if some of this silver solution be dropped into water containing a chloride, a curdy white precipitate of chloride of silver will be formed. If there is also present in the water enough potassium chromate to give a yellow colour, the white precipitate will continue to form as before, owing to the silver having a greater affinity for chlorine than for the chromic acid in the chromate. But, at the moment when all the chlorine in the sample has been converted, the silver will attack the yellow potassium chromate, and chromate of silver will be formed, which is red in colour. The amount of chlorine present is, therefore, shown by the amount of silver solution required to convert it all to silver chloride, and the exact point when the chloride precipitate ceases to form is shown when the chromate indicator turns from yellow to red.

It is not necessary to add the silver solution until the colour becomes very red, as the delicacy of the reaction would be destroyed, but the change from yellow to yellowish-red must be distinct, and must not change on shaking. The sample of water to be tested should be neutral, as free acids dissolve the silver chromate. If it should be acid, neutralise by adding sodium carbonate. Slight alkalinity does not interfere with the reaction, but should the sample be very alkaline, it may be neutralised by nitric acid.

Should it happen that the colour does not change within the limits of the graduations, the sample may be tested by diluting with distilled water. For example, add three parts of distilled water to one part of the sample. If then, on testing the mixture, the colour changes at 200, the number of grains per gallon in the original sample will then be four times this reading, or 800 grains.

The chlorine should be kept down to the least possible amount—say, below 100 grains per gallon—as the nearer the boiler water is kept to that of fresh water the safer the boilers are against corrosion.

EXAMPLES.—A boiler containing 500 gallons of distilled water which has been impregnated with 500 ozs. of sea water (1 oz. to the gallon = 266 grains of chloride per gallon) would, according to the following table, require little more than 3·3 lbs. of lime, or 18·6 lbs. of soda, in order to make it neutral, and therefore non-corrosive. (The exact figures would be: lime, 3·5 lbs.; soda, 19·6 lbs.) Whereas, a boiler containing 500 gallons of water taken direct from the sea would require 17·5 lbs. of lime, or 98 lbs. of soda, in order to bring about the same result.

While the amounts of lime or soda in the first case are well within practical limits, and may be used in the boiler with advantage, those given for pure sea water might cause serious trouble.

It must be clearly understood, that the following table is only given for the engineer's information. It should not be taken as an instruction to be implicitly followed for the amount of soda or lime to be used with various densities of sea water. The engineer can easily estimate the amounts of

soda or lime he is putting in from time to time and the amount of make-up feed that is being introduced into the boiler, by carefully following out the instructions given in the previous "Notes on the Corrosion of Boilers."

TABLE SHOWING THE AMOUNT IN POUNDS OF LIME OR SODA REQUIRED TO COUNTERACT THE CORROSIVE EFFECT WHICH VARIOUS ADMIXTURES OF SEA WATER WOULD HAVE IN 500 GALLS. (5,000 LBS.) OF BOILER WATER.

Lime in Lbs. for every 500 Galls. Boiler Water.	Chlorine Tests in Grains per Gall.	Soda Crystals in Lbs. for every 500 Galls Boiler Water.	Lime in Lbs. for every 500 Galls Boiler Water.	Chlorine Tests in Grains per Gall.	Soda Crystals in Lbs. for every 500 Galls. Boiler Water.
18·48	1,400	104·16	9·24	700	52·08
17·82	1,350	100·44	8·58	650	48·36
17·16	1,300	96·72	7·92	600	44·64
16·50	1,250	93·00	7·26	550	40·92
15·84	1,200	89·28	6·60	500	37·20
15·18	1,150	85·56	5·94	450	33·48
14·52	1,100	81·84	5·28	400	29·76
13·86	1,050	78·12	4·62	350	26·04
13·20	1,000	74·40	3·96	300	22·32
12·54	950	70·68	3·30	250	18·60
11·88	900	66·96	2·64	200	14·88
11·22	850	63·24	1·98	150	11·16
10·56	800	59·52	1·32	100	7·44
9·90	750	55·80	·66	50	3·72

Lecture XVIII.—Questions.

1. Enumerate the different classes of marine boilers at present in general use, and describe briefly their distinctive features. Give freehand sketches of each of these types, and state the maximum pressures at which they are worked. Why has the rectangular boiler been given up, and what two forms of rectangular boilers were in use 50 years ago?

2. Describe the construction of a marine boiler with four furnaces of modern type for high pressure steam. Sketch a cross and a longitudinal section, showing the water spaces, with a complete index of the various parts. How are the flat surfaces stayed? Enumerate all the principal fittings.

3. Describe the construction of, and sketch both in transverse and longitudinal section, a *marine* boiler. Mention some of the causes to which a loss of heat may be attributed when the boiler is in operation.

4. Sketch and describe by an index of parts a cylindrical high-pressure marine boiler with two furnaces, showing the mode of construction and staying, and describe the several processes employed in its construction from the commencement until its completion in the shop.

5. Sketch and describe clearly how the furnace tube of a cylindrical marine boiler is constructed, and how it is attached to the combustion chamber and front end plates, and also how expansion is allowed for.

6. Give a freehand sketch of a marine engine boiler, with all the necessary fittings in their relative positions; name them and their respective uses.

7. Referring to the description of the engines and boilers of the S.S. "Inchdune," sketch and explain in detail the boilers, superheaters, and induced draught arrangements. Point out wherein these are novel or beyond ordinary current practice.

8. Illustrate and describe any method of induced draught for marine boilers. How may the efficiency of a boiler be increased? Why should the feed-water be heated and purified before it is admitted to a high-pressure marine boiler?

9. Sketch and describe by an index of parts Shanks' small vertical marine boiler for steam launches or small tugs.

10. Give a table of analysis of sea water, and calculate the percentage weights of solid matter by reckoning the specific gravity of salt water as 1·026.

11. Mention the chief causes of corrosion in marine boilers, and how these may be prevented.

12. Explain clearly, by aid of a sketch and table, how you would test the proportion of chlorine and total solid matter in feed-waters. What is meant by saying that the saltness of water is $\frac{1}{32}$ or $\frac{2}{32}$ or $\frac{3}{32}$? and show how this curious fraction is used as a standard.

13. Describe in detail the method of testing water for its corrosive properties. State how you could best neutralise these evil properties in feed-water by an example.

14. Describe the chief methods of producing a forced boiler draught, and point out the advantages and disadvantages of each.

LECTURE XIX.

WATER-TUBE MARINE BOILERS.

CONTENTS.—The Navy Boiler Question and the Decision of the Special Committee—Water-tube Boilers—Belleville Boiler—The Babcock & Wilcox Marine Boiler—The Yarrow Small and Large Tube Types—Normand Boiler—Clyde Water-tube Boiler—Thornycroft Boilers of the "Speedy" and "Daring" Types—Comparative Trials of Water-Tube Boilers—Questions.

The Navy Boiler Question and the Decision of the Special Committee.—The vexed problem of the best class and kind of boilers for H.M. Navy occupied the attention of a Select Committee for nearly four years, from 1900 to 1904. They devoted a large amount of care and attention to the arduous task entrusted to them, and conducted exhaustive experiments on board several vessels of the Royal Navy. They brought all their expert knowledge to bear upon the results of their own experiments, as well as upon the evidence and circumstances laid before them.

On August 2, 1904, the "Final Report of the Committee" was issued. They find, that water-tube boilers are essential, and that the Yarrow large tube, together with the Babcock & Wilcox boilers, are the best up to the present, for all large ships, without even the assistance of cylindrical boilers. This is a most important change in the attitude of the said committee, for, in their previous reports, they expressed a preference for one-fifth of the boiler power in these large cruisers and battleships being of the cylindrical or Scotch marine system, and with four-fifths of the water-tube type.

So many difficulties had arisen previous to the appointment of the committee, both inside and outside the British Navy, in regard to the working of the Belleville boiler, and so many accidents had originated with them, that the demand for their abandonment came from many quarters, although with the more recent and systematic system of firing they did very good work. Further, the almost unanimous decision, that the best days of cylindrical boilers for such vessels were past, caused the committee to concentrate their attention and trials chiefly upon the Babcock & Wilcox, the Niclausse, the Durr, and the Yarrow large tube kinds, with the result that they were satisfied

with the first and the last of these four types of water-tube boilers.

In the Babcock & Wilcox boiler the tubes wherein steam is generated are nearly horizontal; whereas, in the Yarrow boiler they are nearly vertical. The committee say, that each of these two types has its own particular advantages, and that it can only be determined by long experience on general service which is the better boiler for naval requirements. They admit, however, that in these two types the resources of engineering are by no means exhausted, and they bespeak an ample trial of any other type of boiler which may in the future seem to possess great merits, with the suggestion, that it should be first of all fitted and tried in a smaller vessel or in a second-class cruiser.

In regard to small vessels of high speed the committee state, that from the nature of the case some form of "express" boiler with small tubes closely pitched is absolutely necessary in order to obtain such a ratio of output to weight of boiler as is now required for torpedo boats and destroyers. For small cruisers which have to remain long at sea and act with the fleet, it is probable that a boiler of the Yarrow large tube type would give better results than the "express" type hitherto adopted.

The Durr boiler was the only one tried which was fitted with superheaters, but the results obtained from the same were not sufficiently convincing to enable them to express a decided opinion on the value of superheating as applied to naval boilers.

We shall now devote a few pages with illustrations to a detailed description of a few of the best known and most approved types of water-tube marine boilers.

Water-Tube Boilers.—Although this type of boiler has been the subject of many patents during the last seventy years, and has been frequently employed for land purposes within the last twenty years, yet it is only quite recently that they have been successfully applied to steamers, and more especially to war vessels of our own and foreign navies. The main objects in view in their adoption as marine boilers have been :—

(1) The reduction in weight of the engines, owing to the higher pressure of steam obtainable, and also of the weight of the boilers.

(2) The raising of steam or the pressure as quickly as possible.

(3) Reducing the quantity of water in the boiler, and hence the dead weight carried.

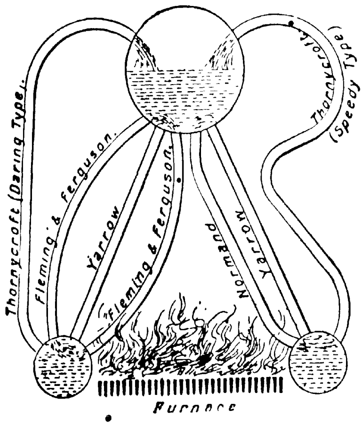
(4) Reducing the risk of injury to firemen; for, should a tube burst, the liberated steam can at once escape up the funnel.

(5) The withstanding of sudden variations in temperature, due to raising or letting down the steam quickly, or leaving the fire doors open.

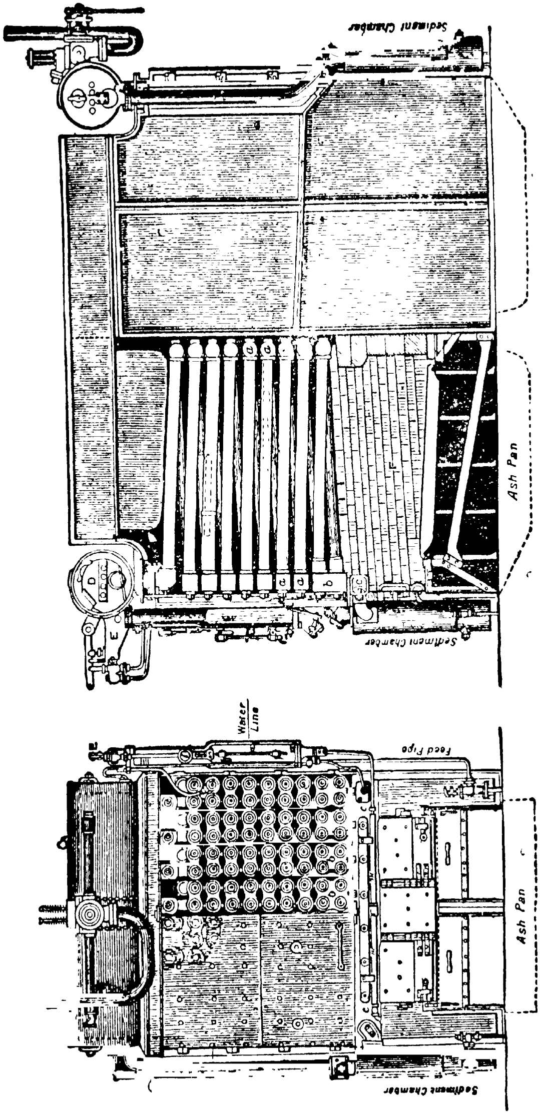
Steam can be raised within half an hour from cold water to a pressure of 200 lbs. in some of the best water-tube boilers; whereas, it not unfrequently **takes from three to ten hours** to get up steam of the required pressure in **the large cylindrical** marine boilers of the type illustrated in Lecture **XXIX. Also, the steam can** be let down very quickly by **withdrawing the fires and** leaving the fire-doors open without causing any leakage or apparent damage to the boiler. Such treatment in the case of large cylindrical boilers would most assuredly result in seriously damaging them. In a well-designed water-tube boiler the water circulates very rapidly throughout the whole system, and the temperature is thereby maintained more uniform than in the cylindrical boiler, where the water under the furnace flues is often quite cold, unless a hydro-kineter or circulating pump is used. This sluggish circulation and consequent difference in temperature between the upper and lower parts of the cylindrical marine boiler produces severe stresses owing to unequal expansion of the plates, and often causes leakage at the furnace mouths and shell-joints.

Water-tube boilers may be divided into two main classes. (1) Those possessing comparatively large tubes (of 3 ins. or more in diameter), inclined less than 30° from the horizontal. The Belleville and Babcock-Wilcox boilers belong to this class. (2) Those having small tubes (less than 2 ins. diameter), and inclined more than 30° from the horizontal, of which the Thornycroft and Yarrow boilers are examples. These two classes may again be subdivided into (*a*) priming or foaming boilers, where the combined water and steam are delivered *above* the water line into the upper or steam drum, as in the Belleville and Thornycroft boilers; and (*b*) drowned tube boilers, where the water and steam are delivered into the steam drum *below* its water level, as in the Babcock-Wilcox, Normand, and Yarrow boilers.

The following skeleton figure will serve to illustrate class (2) and its subdivisions (*a*) and (*b*), more effectually than any worded description, while



SKELETON DIAGRAM OF THE "DROWNED" AND "FOAMING" TYPES OF THE SMALL "TUBE" CLASS OF WATER-TUBE BOILERS.



BELLEVILLE BOILER.

class (1) will be readily understood from the following illustrations of the Belleville and Babcock-Wilcox boilers.

Belleville Boiler.—This boiler belongs to the first or large tube class and to sub-division (*a*), since it is a priming boiler. It consists of from eight to ten sets of tubes at an inclination of 1 in 25, and placed side by side over the fire, together with a feed collector, a steam drum, and a mud drum. Each element is in the form of a flattened spiral of straight tubes screwed into junction boxes of malleable cast iron. These junction boxes are placed vertically above each other with the upper end of one tube on a level with the lower end of the next one. Holes provided with doors and cross bars are fitted to the front boxes for the inspection and cleaning of the tubes. The diameter of each tube is usually about 4½ ins. The lowest junction box of each element is connected to a horizontal cross tube (called the feed collector) at the front of the boiler. The uppermost tube is connected to the lower part of the cylindrical steam drum, which is outside the boiler casing. A vertical circulating pipe or downcomer is also placed outside the casing, and connects the bottom of the steam drum with the mud drum. The upper end of the mud drum is attached to the feed collector. The water line is about halfway up the elements. The feed water, which is delivered into the steam drum at the end furthest from the downcomer, passes along the bottom of the same, and then down the external downcomer, through the mud drum, into the feed collector. It then enters the elements, to be heated by the fire on its way upwards to the steam drum, where it is delivered as a mixture of water and steam. The water and steam are separated by dash plates, and the water, with the addition of any fresh feed, passes along the bottom of the steam drum again to the downcomer, as before. The feed water is supplied by duplex pumps, at a pressure of about 600 lbs. per sq in., and the admission is regulated by an automatic control valve of the float type. Before the feed water is pumped into the boiler it is mixed with lime, in the proportion of about 4 lbs. per 1,000 I.H.P. every twenty-four hours, in order to aid the precipitation of oil and lime salts present in the water. These, on being heated in the steam drum, precipitate in a finely-divided state and settle to the bottom of the mud drum. If they did not do so, they would adhere to the inside of the heating tubes and cause overheating by their forming a non-conducting lining, and thus prevent the due absorption of the heat by the water. The steam, after leaving the boiler, passes through a separator (which is fitted with an automatic valve drain trap), then through a reducing valve to the engine. The effect of this reducing valve is to slightly superheat the steam and ensure a supply of dry steam at a constant pressure to the engines. As long as the pressure in the boilers does not drop below that for which the valve is set, the engines receive steam at a constant pressure.

Another feature of the Belleville system is the admission of jets of air at a pressure of 8 to 10 lbs. per sq. in. above the fire grate, with the object of thoroughly mixing the gases and ensuring complete combustion of the fuel. The gases are caused to pass backwards and forwards among the tubes on their way to the chimney by baffle plates.

The Babcock and Wilcox Marine Boiler *(See the Facing Plate)*.—This boiler consists of inclined tubes forming the bulk of the heating surface, a horizontal steam and water drum, and a mud drum. The tubes, which are of seamless steel, are expanded at both ends into wrought-steel boxes or headers, and thus form vertical sections. By means of connections with the steam and water drum at the upper ends of these headers, the steam generated in the tubes is liberated and water supplied to take its place. The furnace is underneath the nest of tubes, and the gases, as shown by the direction of the arrows, come into intimate contact with all the heating surface. The furnace is lined with firebricks in the case of boilers for the Mercantile Marine, but, for Navy purposes (and in all cases where light weight is of importance), with fire-tiles bolted to the side plates. The whole boiler is enclosed in a special arrangement of iron casing, fitted with fire refractory material, which is very effective in preventing radiation of heat. The steam and water drum is of large volume, and made of wrought-steel plates. The sinuous headers and mud drum are made of wrought steel, and are of such ample strength, that, no stay-bolts are required, with even the highest pressures. Opposite the end of each tube, or group of tubes, is an internal fitting or door, of oval or square shape. The joint is made on the inside of the header by means of an asbestos wire-woven ring The door is drawn up into place by an outside bolt and nut, and dog, or cap. All the steam mountings, such as stop and safety valves, feed-check valves, water and pressure gauges and scum valves, are attached to the steam and water drum; the blow-out valves are attached to the mud drum. The steam and water drum is fitted with wash plates to prevent undue motion of the water when the ship is rolling.

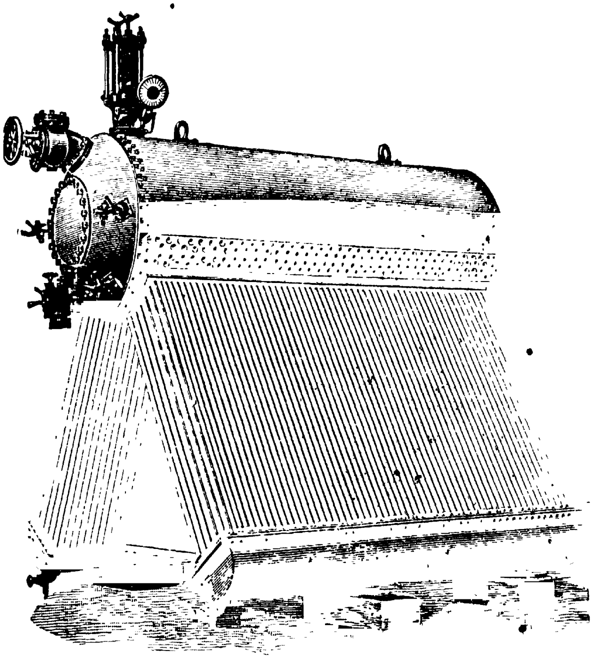
The steam generated in the tubes rises vertically through the rear headers into the steam and water drum, whence the water returns to the front headers and the tubes. Thus, there is a continuous circulation of water in one direction, and this continuous circulation gives water of an equal temperature in all the parts, so that undue stresses from unequal temperatures are avoided. The interior of the boiler offers the greatest facility for cleaning, and, by means of doors arranged in the side casings, there is equal facility for the removal of soot. There is a firebrick roof to the furnace, extending from the front headers towards the rear, for about two-thirds of the length of the tubes between the headers. From the rear end of this furnace roof baffle plates of iron rise vertically, and direct the heated gases upwards. These plates reach to about two-thirds the height of the tubes in a section. The gases pass over the top of these plates, and are directed downwards by horizontally-inclined baffles (resting on the top row of the inclined tubes) and a second set of vertical baffles, as shown in the illustration. The gases pass under the lower edge of the second

set of baffle plates, rise up between this and the front header, and pass by the steam drum among the two top rows of return tubes and into the uptakes. *(See the wave line on the Longitudinal Section of the facing Plate.)*

The feed-water enters the steam drum below the water level, passing vertically through a contracted nozzle into the steam space, and is deflected downwards by a cover or guard into the water space, the air being liberated before the feed-water enters into the circulation. The greater part of the suspended matter in the feed water is deposited in the mud drum. The water in the tubes is heated by the direct heat of the furnace gases while passing through the inclined tubes. Boiling water and the steam thus formed then rise in the rear headers, and pass through the return tubes into the steam and water drum. The water is deflected downwards by baffle plates, and the steam escapes through a dry pipe in the top of the steam and water drum to the main steam stop valve.

The Yarrow Boiler.—This kind of boiler belongs to the small tube type. It is, in general form, a triangular prism, the steam drum forming the apex, and the two lower drums with the fire grate, the base. It differs from most of the small tube boilers in having straight tubes, which are expanded into the drums.

In the smaller sizes these drums are in halves and bolted together, but in the larger sizes the steam drum is cylindrical and riveted, the lower chambers being semi-cylindrical or ◡ shaped, with a flange to which the tube plate is bolted.

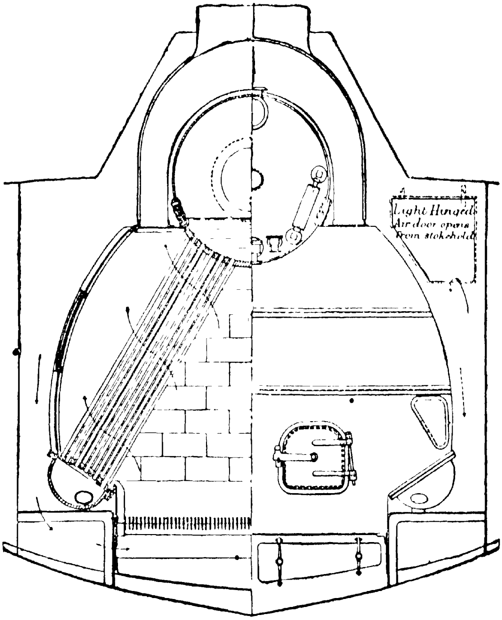


PERSPECTIVE VIEW OF YARROW BOILER WITHOUT CASING AND FIRE GRATE.

As first designed, these boilers had the drums carried outside the casing and downcomers fitted to permit the water to circulate from the upper to the lower drums, and thence up through the tubes again; but in the more recent designs, these downcomers have been discarded. Mr. Yarrow has made numerous experiments, which have satisfied him that these are unnecessary. One of these consists of a model of a section of his boiler made with two glass tubes. He applies heat to one of these tubes, thus causing a circulation of water up the heated tube and down the cold one.

On applying heat also to the cold tube, the circulation becomes more rapid. He then removes the source of heat from the up-take tube, still applying heat to the down-take one, and the circulation continues as before. This proves, at least, that if circulation is started it is not easily stopped; but there is no doubt that the ordinary tubes act alternately as *up* and *down* comers respectively.

The tubes are expanded into the tube plates and steam drum. These were originally made of copper, but at the high temperatures and pressures now in use, copper is unreliable, and besides which it is difficult to



YARROW BOILER.

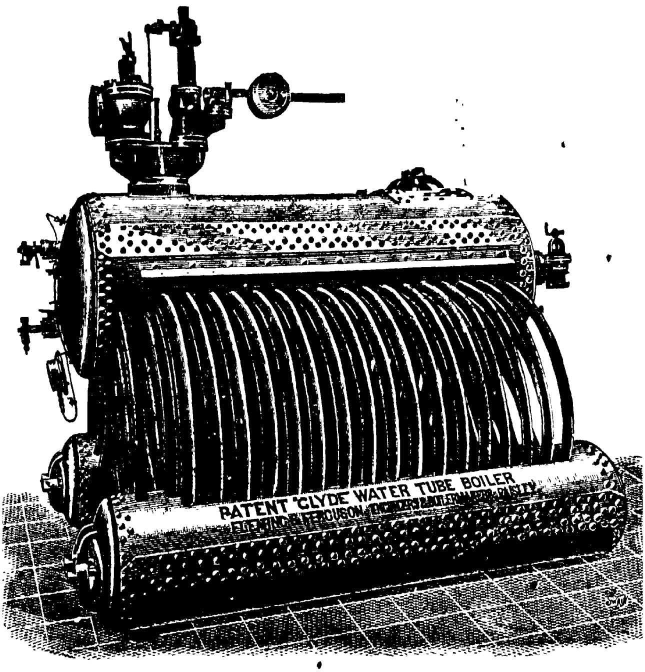
HALF CROSS SECTION. HALF END VIEW.

detect a flaw in a newly drawn copper tube, consequently *steel* tubes have been substituted and found quite satisfactory. The tubes in the Yarrow boilers for the Dutch Government are $1\frac{1}{8}$ in. in diameter by 5 ft. long, and are seldom fitted of less diameter than 1 in.

The Yarrow boiler of H.M.S. "Hornet," which has a heating surface of 1,027 sq. ft., and a grate surface of 20·6 sq. ft., and weighs, with its fittings and water complete, 5·35 tons, was found to evaporate 12,500 lbs. of water from 60° F. to 180 lbs. pressure per hour. In this case, the tubes were of copper and 1 in. in diameter. Length of fire bars and steam drum were $6\frac{1}{2}$ ft. and $7\frac{1}{2}$ ft. respectively, and steam could be raised to full pressure in twenty minutes from lighting of fires.

Normand Boiler.—The Normand boiler is somewhat similar to the "Yarrow," but it has outside downcomers and curved tubes, as indicated by the skeleton diagram at the beginning of this article. This construction renders it less liable to stresses during the rapid raising of steam, the forcing when at work, and the sudden cooling to which water-tube boilers are often subjected; but on the other hand, should the necessity for the removal of a tube arise, many of the surrounding tubes have also to be displaced.

Clyde Water-Tube Boiler.—This boiler, which is manufactured by Messrs. Fleming & Ferguson, of Paisley, is shown in perspective with the outer casing removed, in following figure. Recently two of these boilers



THE "CLYDE" WATER-TUBE BOILER.

were fitted into the Canadian cruiser "Aberdeen," and they have been reported upon favourably by the chief engineer.

This boiler is also of the drowned tube type, and resembles the Yarrow in form, but the tubes are curved, and are so arranged that any one of

them may be readily removed by withdrawing them into the upper drum. The diameter of the upper drum is 6 ft., and that of the two lower drums, 3 ft. The tubes are expanded into the top and bottom drums in the ordinary way, and are $2\frac{1}{4}$ ins. diameter, except for a short length at the upper ends, where their diameter is increased to $2\frac{5}{16}$ ins. They are placed zig-zag, so that the flame from the furnace has to wind through them.

Thornycroft Boiler.—Two forms of this boiler are shown by the following figures, and are known as the "*Speedy*" and "*Daring*" types respectively.

The Thornycroft is one of the most efficient forms of water-tube boilers, but is open to the same objections as the Normand—viz., the use of curved tubes, which involve considerable trouble in their removal or replacement, examination, and cleaning. The tubes connecting the lower and upper drums enter the latter above the water level, thus constituting a boiler of the foaming type. The jets of water and steam from the tubes impinge against baffle plates, which cause the water to drop down into the steam drum, and allow the steam to pass freely without having to force its way through the water, as in the case of the drowned tubes in the Yarrow and Normand boilers.

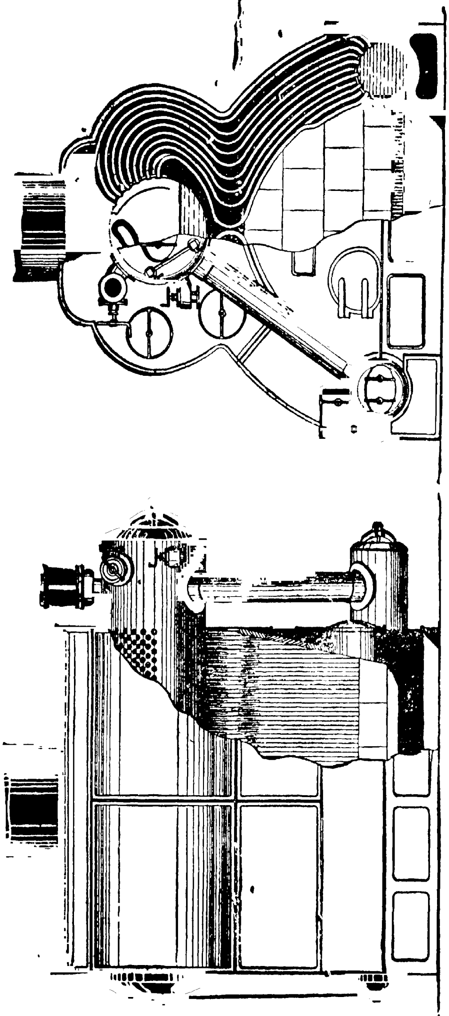
In trials for economy, the Thornycroft boiler has evaporated as much as 13·4 lbs. of water from and at 212° F. per lb. of coal when burning 7 lbs. of coal per sq. ft. of grate per hour.

The earlier, or "Speedy" type, has external downcomers. The newer, or "Daring" type, has the downcomers between the steam drum and the centre water drum, and a connection outside between the back ends of the centre water drum and wing drums, as shown at the extreme right-hand of the semi-cross sectional figure. The outer casing is made of very thin galvanised sheet iron, with an inner thick lining of asbestos millboard, which thus prevents radiation and damaging of the casing

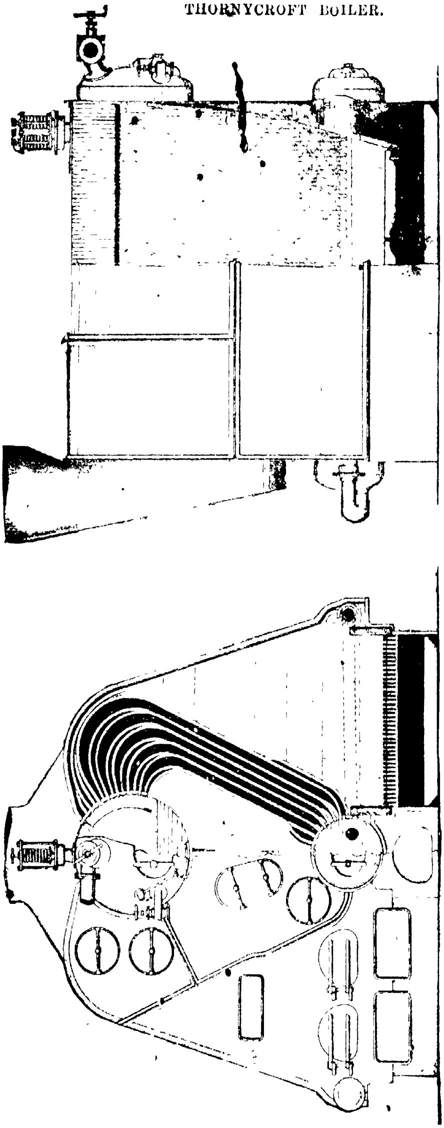
One advantage of the Thornycroft boiler, and particularly of the later type, is the large combustion chamber, which allows thorough mixing and combustion of the gases, and must therefore greatly conduce to the economy of the boiler.

I.—Comparative Table of Trials of Water-Tube Boilers

Type of Boiler.	Ship or Trial	I H P per ton of Boiler, Water and Fittings	Water Evaporated per lb of Coal	Coal Burnt per sq ft. of Grate.	I H P. per sq ft of Grate	I H P per sq ft. of Heating Surface	Lbs Coal per I H P hour	Weight of Boiler per sq ft of H S in lbs	H S / G S.	Boiler Pressures.
			lbs	lbs						
Belleville	"Kherson"	28	..	..	11·7	·377		26 3	31 2	.
"	"Powerful"	..	.	16·1	.	.	1 8		.	.
"	Barrow Coy.	..	10 81	18	..		.	..	31 5	260
"	"	..	10 42	24	.	.	.	.	31 5	260
Babcock & Wilcox	"Zenith City"	21 2	7 05	25·9	11 5	·275	2 25	25 5	41 8	190
	U.S.N. Test	..	8·4	44·1	.		.	31 7	40 3	170
Yarrow	Dutch Govt.	109	.	..	21 6	·43		8 9	50	
"	"Hornet"	81·5	..	..	22	·427	..	11 6	50	180
Thornycroft	..	..	13·4	7·7	..	.	.	..	61	..
"	..	..	12·0	29·8	15	·245	1·99	..	70	..
"	"Desperate"	..	10 29	66·8	29·5	422	2·26	..	50	..



THORNYCROFT BOILER—KNOWN AS THE "SPEEDY" TYPE.



Improved Thornycroft Boiler—Known as the "Daring" Type.

II.—Comparative Trials of Water-Tube Boilers.

Name of Steamer.	Boiler Pressure.	Indicated Horse-Power	Air Pressure in inches of Water.	Number and Type of Boiler.	Total Grate Surface	Total Heating Surface.	Total Weight of Machinery	Total Weight of Boilers	I H P per ton of Machinery.		I H P. per ton of Boilers		I.H P per sq. ft. of Grate Surface		Heating Surface per I H P	
									Natural Draught	Forced Draught	Natural Draught	Forced Draught	Natural Draught	Forced Draught	Natural Draught	Forced Draught
					sq ft	sq ft	tons	tons								
"Majestic" . .	155	12,497	0·77	8 Cylindrical .	821	25,570	1,300	736	8·01	9·61	14·15	16·99	12·68	15·22	2·45	2·04
"Powerful" . .	260	25,000	..	48 Belleville .	2,192	69,453	2,250	1,184	8·0	11·11	15·2	21·1	8·21	11·40	[illegible]	2·77
"Sturgeon" . .	200	4,492	2·5	4 Blechynden .	156	9,652	109	605	..	41 21	..	74·24	..	28·79	..	2·14
"Avon" . . .	250	6,000	2·5	4 Thornycroft .	236	13,400	145	81	..	41·31	..	74·07	..	25·42	..	2·23
"Terrible" . .	260	25,000	..	48 Belleville .	2,260	67,800	..	..	..	..	..	..	..	..	..	..
"Kherson" . .	250	13,307	..	24 Belleville .	1,132	35,352	..	475	..	..	..	28	..	11·7	..	2·65
"Ohio" . . .	250	1,500	..	4 Belleville .	192	6,013	..	..	..	..	.	..	..	..	..	..
"Renown" . .	150	1,290	..	8 Single ended return tube	1,132	..	..	584	..	..	.	24	..	..	..	..
"Zenith City" .	190	1,540	..	Babcock-Wilcox	67	2,800 and 600 in heater	..	..	..	..	..	21·2	..	11·5	..	1·9

LECTURE XIX.—QUESTIONS.

1. Give a concise clear idea of the 1904 finding of the Special Committee on the Navy boiler question.

2. State the main objects to be kept in view in the adoption of water-tube boilers for steamships. Illustrate by diagrammatic sketches two main classes of water-tube boilers.

3. Sketch and describe in detail the Belleville boiler. State its good and bad qualities.

4. Sketch and describe in detail the Babcock & Wilcox marine boiler. Point out its special features and qualifications.

5. Compare by aid of sketches, with index to parts, the small and the large tube Yarrow type of boiler. State their respective qualifications for different types of vessels.

6. Sketch and describe in detail the "Speedy" and "Daring" types of Thornycroft boilers. Compare their qualifications and special uses.

7. Describe, with sketches, some one form of water-tube marine boiler with which you are acquainted. No fittings need be shown. What are the most important things to remember when designing these boilers?

8. Describe, with the aid of a sketch, any type of reducing valve for reducing the pressure of steam from a higher variable to a lower constant pressure, explaining clearly its action and how the lower pressure may be adjusted. If the pressure on the boiler side of the valve be 300 lbs. per square inch gauge (corresponding temperature 422° F.) and on the engine side be 250 lbs. gauge (corresponding temperature 406° F.), calculate the dryness fraction on the engine side if that on the boiler side be ·95.

9. Describe, with the aid of sketches, the construction of one type of water-tube boiler as used in the Navy, and point out the special advantages of this type of boiler for naval purposes.

10. Describe, with as many detail sketches as you consider necessary, one of the following:—(a) A locomotive boiler; (b) a Belleville boiler; (c) a Lancashire boiler. Any staying used must be carefully shown, and the method of calculating it explained, and the feed and steam valve arrangements fully explained.

LECTURE XX.

BOILER CONSTRUCTION.

CONTENTS.—Materials used in Boiler Construction—Wrought-Iron, Steel, Copper—Joints of Boiler Plates, Riveted Joints, Punching and Drilling, Hand and Machine Riveting, Caulking, Welded Joints—Methods of Connecting the Parts of the Shell, and Flues—Staying of Boilers—Strength of Boiler Shells—Strength of Flues—Strengthening Hoops for Flues—Corrugated Furnaces—Questions.

Materials used in Boiler Construction.—The earliest forms of steam boilers were constructed chiefly of cast-iron, but on account of the low tensile strength of this material and its unreliable nature when subjected to the variable temperature and stresses in a steam boiler it has been abandoned for many years.

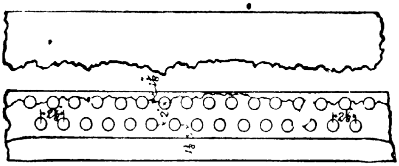
Wrought-Iron.—Until the recent introduction of mild, soft steel, wrought-iron was the material which was almost exclusively employed for the construction of steam boilers. Wrought-iron possesses great tenacity, combined with the important qualities of toughness and ductility. It is therefore well adapted to resist sudden strains and alterations of temperature, and does not give way suddenly or without warning. Also, its capability of being welded, forged, and flanged, adds to its value as a material for boiler construction; whilst it is a matter of importance that its strength is not influenced to any appreciable extent by a moderately large increase in temperature, such as high-pressure boilers are subjected to under ordinary circumstances.

The average ultimate tensile strength of wrought-iron bars may be taken at 22 tons (or about 50,000 lbs.) per square inch; the tensile strength of the best quality of bar iron being about 25 tons (56,000 lbs.) per square inch. The average tensile strength of wrought-iron plates as used for boilers, is—

With the grain = 21 tons (47,040 lbs.) per square inch.
Across „ „ = 19 „ (42,560 „) „ „ „

The plates used in boilermaking should all be of good quality; inferior plates give great trouble and are always unsatisfactory. The plates require to pass through some of the various processes of flanging, dishing, welding, punching and rolling cold and in

dealing with inferior plates the greatest care must be exercised by the workmen to prevent them from injury before they are fitted into the boiler.



Fractured longitudinal joint in a new boiler from the use of a brittle iron plate. Norwich explosion, 25th September, 1866. The plates were of Cleveland iron, and when tested by bending, broke off short. They were quite wanting in ductility.

Only a very good quality of plate will stand flanging with impunity; and where joints have to be welded, satisfactory work cannot be obtained with an inferior plate which is wanting in ductility. Many inferior plates have a high tensile strength, but are brittle and do not possess that toughness and ductility which are essential qualities when much forging has to be done. The plates of the furnace flues are the most important, since these are more severely taxed by variations of temperature (causing sudden expansions and contractions), than any other plates in the boiler. The plates of the furnace crowns are alternately in contact with the fierce hot flames from the fire, and the currents of cold air which rush into the furnace each time the firing door is opened. The constant straining of the plates which is thus induced is very trying, and none but plates of very good quality will stand it for a great length of time. The various brands put upon plates, such as *Best*, and *Best Best*, &c., are very misleading, the "treble Best" plates of one maker being no better in some cases than the "Best" plates of another. It is only by the use of efficient testing machines, careful chemical and microscopical analysis, that a thorough knowledge of the capabilities and nature of a given quality of plate can be ascertained.

Steel.—This material has come into use very largely for boiler plates within the last few years, owing to the valuable properties it possesses when manufactured in the mild form. Mild steel boiler plates containing about 0·1 per cent. of carbon are now manufactured by the Bessemer, Siemens and basic (Thomas-Gilchrist) processes, and have an ultimate tensile strength of from 25 to 30 tons per square inch, with an elongation in test strips (8 inches long) of from 20 to 25 per cent. Steel plates with a

higher tensile strength than this, are usually too hard and brittle for boilermaking purposes. Owing to the greater tensile strength of steel, boilers made of that material are much lighter than when made of iron, and the plates being thinner, the joints are more easily made tight. Good mild boiler steel plates also possess a ductility superior to wrought-iron, and are therefore more suitable where flanging has to be done. They can be treated whether cold or hot by experienced workmen with the same freedom and usage as applied to malleable iron plates, except to a small extent in the case of welding, for the steel plates do not weld quite so freely as iron ones, and the welds are not so trustworthy. Steel scrap, however, welds into blooms quite as freely as wrought-iron scrap, and the forgings are generally superior.

There have been a few cases of the failure of boilers constructed of steel, the plates of which had been tested before they were used and found to be of a good quality. These failures have engendered in the minds of some engineers a certain amount of distrust of this material. Steel plates are undoubtedly more severely injured by punching than iron plates, and should always be annealed afterwards. Much of the distrust which has been felt in the use of steel for boilermaking has been caused by the use of plates quite unsuitable for that purpose, and their subsequent failure. Engineers have in many cases been too anxious to avail themselves of the high tensile strength of steel, forgetting that in so doing they are sacrificing the all-important quality of ductility. Steel is undoubtedly superior to iron as a material of construction for steam boilers, but great attention and care must be paid to its special properties and the methods of manipulation most suitable for it. Plates of a very mild nature, possessing moderately high tensile strength but great ductility, should invariably be used. Of all the tests applied to steel plates the bending test is the most valuable.

Copper.—Occasionally, copper has been used for boilermaking, although chiefly for small boilers. It is a much better conductor of heat than iron or steel, the ratio of the conductivities of copper and iron being expressed approximately by the numbers 74 to 12. It wears better under the intense heat of the furnace, and gives a higher evaporative efficiency. It has also the advantage

* See paper on "Injurious Effects of a Blue heat on Iron and Steel" by Mr. Stromeyer, read before the Institution of Civil Engineers. Vol. lxxxiv. of Proceedings. See also "General Remarks on Steel Boilers" by Thomas Trail, in his Pocket-Book on Boilers, Published by Charles Griffin & Co., London.

of resisting oxidation, or corrosion from the feed water. It is very ductile and malleable, and can therefore be worked with great ease, and will stand a considerable amount of straining action. It has, however, one great disadvantage—viz., its strength decreases to a large extent with an increase of temperature. At 32° Fah. its tensile strength is, on the average, 15 tons per square inch, but at 550° Fah. its tensile strength is reduced to about 75 per cent., and at 850° Fah. to 50 per cent. of this value. On account of this inferior strength and the high price of copper, its use for boiler making has been entirely given up, except for locomotive fire-boxes and stays.

Joints of Boiler Plates.—The joints in boiler shells and flues are formed either by riveting or welding.

Riveted Joints.—These are of various forms and strengths, but they may all be classified into, (1) lap-joints, (2) butt-joints.

The next set of diagrams, Nos. 1 and 2, show a single-riveted lap joint. This is the simplest but least efficient form of joint, and is only employed where great strength is not required.

A joint of this kind may be fractured in four different ways.

(1.) By the shearing of the rivets between the plates.

(2.) By the tearing of the plates along the line of rivet holes.

*(3.) By the crushing of the plate between its edge and the rivet holes, causing the metal between the edge of the plate and the rivet, to be forced out.

(4.) By the breaking of the plate between the rivet hole and the edge, in a line at right angles to the edge.

Another resistance must also be overcome before fracture takes place, viz., the frictional resistance of the plates.† The contraction of the rivets in cooling, compresses the plates so

* The third method of fracture cannot be avoided by giving lap to the plates, *if the surface of the holes* in the plate which takes the pull of the rivet *is too small.* In this case, the surface will be crushed, however large the lap of the plate is. This determines the diameter of the largest rivet that can be used for a given thickness of plate.

† The rivet must not be so small that it is unable to draw the plates firmly together, for if it is, the joint is unsatisfactory, and the rivet is under such great tension that the head is apt to fly off when it cools, or afterwards, when the plate is being caulked. The plates should be pressed so tightly together when riveting takes place, and even until the rivet and the immediately surrounding part of the plate have been cooled, that the frictional resistance between the plates is sufficient to prevent them slipping over each other to the minutest extent due to the maximum stresses likely to occur. Some makers apply cold water to the plates, close to the rivets, when they are being riveted up, in order to secure this desirable object with a saving of time and money.

tightly together, that a considerable frictional resistance is set up, which aids in preventing the plates from sliding over one another. The amount of this friction, however, is entirely dependent on the temperature at which the rivet is put in, and the tightness of the joint when the head is formed, and as it cannot therefore be calculated with any degree of accuracy it is usually neglected, in estimating the strength of a joint.

To find the best proportions for a riveted joint.

Let t = The thickness of the plates in inches.
d = ,, diameter of the rivets ,, ,,
a = ,, area of the rivets in square inches $= \pi r^2 = \frac{\pi}{4} d^2$
p = ,, pitch of the rivets in inches.
S_s = ,, shearing strength of the rivets per square inch.
S_t = ,, tensile ,, ,, ,, plates ,, ,, ,,

Consider the strength of a jointed strip of plate of breadth $= p$. Then the best proportion will be obtained when the tearing resistance of the plates = the shearing resistance of the rivets.

$$(p - d)t \times S_t = a \times S_s$$

The shearing strength of rivet iron is usually about equal to the tensile strength of plate iron, therefore, we have

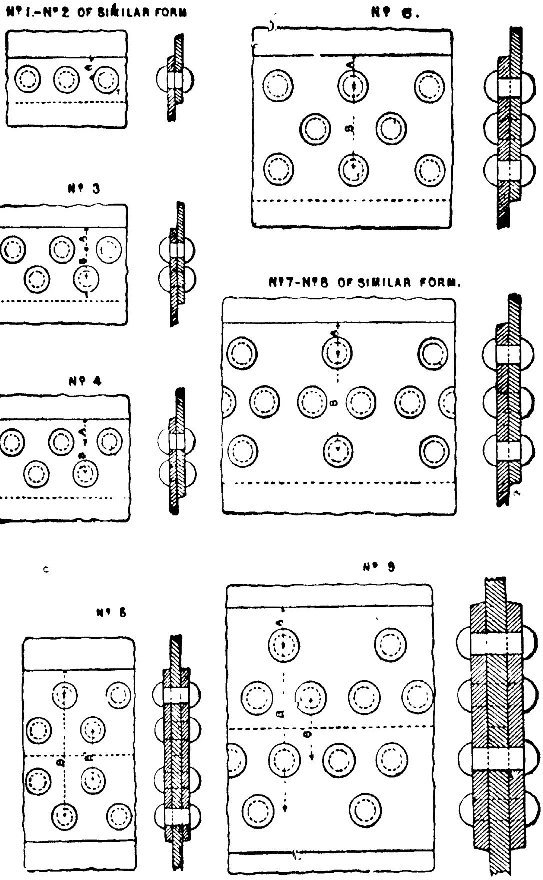
$$(p - d)t = a; \therefore p = \frac{a + dt}{t} = \frac{a}{t} + d.$$

The following relation between the diameter of the rivets and the thickness of the plates is given by Prof. Unwin,

$d = 1{\cdot}2 \sqrt{t}$, which gives a very good proportion.

The following table and sketches (from Smart on "Steam-boilers," see foot-note, next page) show examples of several of the best types of riveted joints in use in modern practice. In calculating the strength of the joints the tensile strength of the steel plates has been taken at 28 tons per square inch; the shearing-stress of the steel rivets at 23 tons per square inch, and that of the iron rivets at 18 tons per square inch. In all the examples shown the shearing stress of the rivets is in excess of the tensile strength of the metal left between the rivet-holes, and it has been found that a considerable excess in this direction adds to the strength of the joint, and at the same time renders it more easily made and kept tight. As the steel plates have all been either drilled, or punched and afterwards annealed, no allowance for reduction of the strength of the metal left between the rivet-holes has been made; nor has any accession of strength been allowed for in those cases in which the holes have been drilled through the solid plates, although the strength of the metal left between the holes, at least in those joints with a moderately close pitch of rivets, will be somewhat increased.

EXAMPLES OF RIVETED JOINTS.



These, as well as several other figures in this lecture, have been supplied by the kind permission of the Council of the Institution of Civil Engineers, from Mr Smart's paper on "Steam Boilers" (see vol. lxxx. of *Proceedings*).

EXAMPLES OF RIVETED JOINTS.

Number of Joint.	Thickness of Plates.	Material of Plates.	How Perforated.	Diameter of Rivet Holes.	Material of Rivets.	Pitch of Outer Rows of Rivets.	Total Breadth of Overlap.	A.	B. and B¹	Thickness of Covering Strips	Tensile Strength of Joint in Percentage of that of Solid Plate.
	Inch.			Inch.		Inches.	Inches	Inches.	Inches.	Inch	
1	$\frac{5}{8}$	Iron	Punched, not annealed	$\frac{13}{16}$	Iron	2	$2\frac{1}{4}$	$1\frac{1}{8}$	.	..	43. Deduced from experiments by Kirkaldy, 1877.
2	$\frac{7}{16}$	,,	,, ,, ,,	,,	,,	,,	,,	,,	...	...	40. ,, ,,
3	$\frac{9}{16}$ full	Steel	Holes punched in outer belts of plates; afterwards enlarged by drilling: the inner belts drilled.	,,	,,	$2\frac{1}{8}$	$3\frac{1}{2}$	$1\frac{1}{16}$	$1\frac{3}{8}$ full	...	68. By calculation.
4	$\frac{5}{8}$	,,	Punched, afterwards annealed.	,,	Steel	$2\frac{1}{2}$	$3\frac{11}{16}$	$1\frac{1}{8}$	$1\frac{5}{16}$	...	68. ,, ,,
5	$\frac{1}{2}$	,,	,, ,, ,,	,,	Iron	$2\frac{7}{8}$	$8\frac{1}{4}$	$1\frac{3}{16}$ scant	B $5\frac{5}{8}$, B¹ $2\frac{1}{16}$	$\frac{3}{8}$	72. ,, ,,
6	$\frac{5}{8}$	,,	Drilled	1	Steel	4	7	$1\frac{1}{2}$	4	...	75. ,, ,,
7	,,	,,	,, ...	,,	,,	5	$7\frac{3}{4}$	$1\frac{1}{4}$	$4\frac{3}{4}$	...	80. ,, ,,
8	$\frac{7}{8}$	,,	,, ...	$1\frac{1}{8}$	,,	$6\frac{7}{8}$	$10\frac{3}{4}$	$2\frac{1}{16}$	$6\frac{5}{8}$	...	80. ,, ,,
9	1	,,	,, ...	$1\frac{1}{4}$	,,	$5\frac{5}{8}$	$12\frac{1}{8}$	2	B. $8\frac{1}{8}$, B¹. $3\frac{1}{8}$	$\frac{13}{16}$	80. ,, ,,

The fracture of the plates by methods (3) and (4) depends upon the distance between the edge of the hole and the edge of the plate. If the lap of the plates be made from 3·2 times the diameter of the rivets when these are less than 1 inch diameter, to 3 times the diameter of the rivets when these are greater than 1 inch diameter, the joint will be equally strong to resist fracture in those two ways. If the lap is made more than this, there is difficulty in caulking since the plate springs.

Lap joints may be single, double, or triple riveted. A double-riveted lap joint is shown at figure No. 3 in the last diagram. In this joint there are two rows of rivets, and their pitch may be found in the same way as before.

Tearing resistance of plates = shearing resistance of rivets.

In this case, a strip of the jointed plate of breadth = p, has two rivets in it.

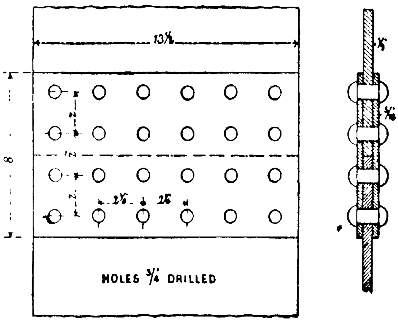
$$\therefore (p - d)\, t \times S_t = 2\, a \times S_s$$
$$\text{or } (p - d)\, t = 2\, a$$
$$\therefore p = \frac{2\, a}{t} + d.$$

When two plates which are lap jointed are subjected to a tensional stress acting at right angles to the joint, the plates tend to draw into line, and bending takes place at points opposite the edges of the overlapping plates. In a lap joint in this position, and subject to varying stress, there is a constant bending and straining motion going on about these points. These points, therefore, become particularly vulnerable to the corrosive action of the water in a boiler, and in the lap joints of a boiler which has been long at work, the inside of the plates is often found to be corroded or eaten away in a line parallel to the joint, and just at the beginning of the lap. This corrosive action, in time, greatly reduces the strength of the plate, especially if impure feed water be used, and it is known as *grooving*, *furrowing* or *guttering*, on account of the deep groove, furrow or gutter which is eaten out of the plate at this special point. The action is due, partly to the mechanical motion of the joint, and partly to the chemical action of the feed water. In lap joints with thick plates, this bending action is greatest. Such joints have therefore a less percentage of the strength of a solid plate when made of thick than of thin plates.

Fig. No. 5 in the last diagram shows a double-riveted *butt* joint. In this joint, the plates are in line, being placed edge to edge, and the connection is made with single, or double-cover plates. These joints are variously made—viz., single, double, and triple riveted.

They have the same proportions as to pitch and diameter of rivets as lap joints. When a single-cover plate is used, the bending action is not altogether avoided, but with double-cover plates there is no tendency to bend, so that this injurious action may be entirely prevented by the use of the butt joint. When double-cover plates are used, the rivets are placed in double shear, *i.e.*, they must be cut through in two planes before the joint can give way. The butt joint with double-cover plates is the most efficient form of riveted joint, but it is also the most expensive. As compared with a lap joint, double the number of rivets require to be put in, and more than double the number of holes have to be punched or drilled.

In practice, double riveting is done in either of two ways. In the last figure referred to, viz., No. 5, the rivets in one row are placed opposite the spaces between the rivets of the other row. This method of riveting is known as *zig-zag* riveting. In the following figure, the method known as *chain* riveting, in which the rivets are placed immediately opposite each other, is shown. Zig-zag riveting requires less lap than chain riveting, and also makes a tighter joint, but the plates are not so strong, especially if the holes are punched. Owing to the greater strength of the chain-riveted joint, it is coming more into use than heretofore.



The following table, copied from Sir John Anderson's *Strength of Materials*, gives the relative strengths possessed by different forms of riveted joints. The strength of the solid plate is taken as 100, and it will be noticed, as we would naturally expect, that even the best of these joints falls far short of the solid plate in strength:—

RIVETED JOINTS.

Description of Joint.	Riveting	Rivet Holes.	Percentage of Strength of the Solid Plate Possessed by the Joint.
Lap, ...	Single, .	Punched, .	55
		Drilled, .	62
Lap, ...	Double, .	Punched, .	69
		Drilled, .	75
Butt, 1 cover, . .	Single, .	Punched, .	55
		Drilled, .	62
Butt, 1 cover, . .	Double, .	Punched, .	69
		Drilled, .	75
Butt, 2 covers, . .	Single, .	Punched, .	57
		Drilled, .	67
Butt, 2 covers, . .	Double, .	Punched, .	72
		Drilled, .	79

The student will see from this table that the single-riveted lap joint is the weakest form; also, that butt joints with single-cover plates, are not any *stronger* than lap joints, but, when double-cover plates are used, the percentage of plate strength is rather greater than that of lap joints.

Single-riveted lap joints are used for the circumferential joints of land boilers, up to about 5 ft. diameter, and working with a steam pressure of not more than 60 lbs. per square inch. Double-riveted lap joints are used for the circumferential joints of marine boilers, and for the longitudinal joints of land boilers of small diameter. Triple riveting, either in the form of lap or of butt joints, is used for the circumferential seams of marine boilers of large diameter and working at high pressures.

Punching and Drilling.—The holes in the plates of riveted joints are either punched or drilled. Each method offers some advantage which is not to be found in the other. The main objection to punching the holes is, that damage is done to the plate by the process. The extent to which a plate suffers from punching depends upon its quality. The injury done by punching to good tough ductile plates is very trifling, but when the plates are of a hard steely nature, their strength may be seriously impaired by the process. Mild steel plates stand punching very well, but, if the plates are at all hard, they may be considerably injured, even to the extent of a loss of tenacity of 30 per cent.

For this reason, it is usual to anneal steel plates which have been punched; after which, they are found to regain their original strength and properties. If the plates are not punched very carefully, it is often found, on putting them together, that the holes do not correspond In order to admit the rivets, and bring the holes as nearly fair as possible, drifting is sometimes resorted to. This reprehensible practice is very injurious to the plates. It consists in driving a round tapered steel pin (known as a "drift") into the hole, in order to remove the obstruction. When the holes do not quite coincide, a drill should be run through them, and, if necessary, a larger rivet used, *but drifting should never be allowed.* In order to obtain perfect agreement of the rivet holes, some boilermakers punch the holes rather less in diameter than the size of the rivet, and when the plates are put together, they are then rimered out to the full size.

When the plates are drilled *separately*, it cannot be said that the holes correspond much better than when the plates are punched, and no advantage in this respect can be claimed for drilled holes. The only way to ensure absolute coincidence of the holes in the different plates, is to have the plates drilled when fixed in position. A number of boiler drilling machines are now in use, which effect this object. The shell plates, after they have been bent, are fixed together by service bolts, and the part of the shell so formed is placed upon a turning-table or some other arrangement for moving the shell round in front of the drills. There are usually two or more drills which operate simultaneously, all round the outside of the boiler shell, and these pierce through two or more thicknesses of plate. When each set of holes has been bored the turning arrangement moves the boiler shell through a distance equal to the pitch of the rivets, and the drills then proceed with the next set of holes. With an efficient machine of this kind the extra expense of drilling over that of punching the holes, is very trifling.

The holes formed by punching are necessarily tapered, since the hole in the die-block is always a little larger than the punch, and when the plates are put together the small ends of the holes are placed inside the joint, and the larger ends outside. By this arrangement, when the rivets are forced into the holes they hold the plates more firmly, and make a much tighter joint, than with the parallel holes of drilled plates; and, even although the rivet heads should be knocked off, the rivets would still retain a firm hold on the plates. The edges of drilled holes are sharp, and exercise a cutting action on the rivet, and it is found that increased strength is obtained by slightly counter-sinking the holes, but this adds considerably to the expense. This cutting

action is not experienced when the holes are punched, for the outer edge of a punched hole is not so sharply defined as a drilled hole. It has been found by experiment that when the plates are punched, the rivets are stronger, but the plates are weakened to a *greater* extent. Hence, as will be seen from the previous table, joints made with drilled holes are rather stronger than those in which the rivet holes have been punched.

Hand and Machine Riveting.—Formerly, all the joints of boilers were riveted by hand, but machine riveting is now used for all the joints to which a machine can be applied. The work done by good riveting machines is much superior in strength to hand work, and can be done much more expeditiously. The hydraulic riveting machine is the one which is most used at the present time, and seems to be the most suitable for the work. In riveting by hand, the blows are so sudden, that the part of the rivet struck by the hammer absorbs nearly the whole energy of the blow, and the formation of a shoulder commences before the hole is properly filled. In machine riveting, on the other hand, the pressure comes gradually on the whole body of rivet, and compresses it fully into the hole before forming a head at all; the joint is therefore much more secure. Before riveting a joint, care should be taken to have the plates drawn closely together, or the compression of the body of the rivet into the hole may cause a slight shoulder to be formed between the plates, and this prevents the closing of the joint.

We have seen from the previous table, that riveted joints are very much weaker than a parallel section of the solid plate, even a double-riveted butt-joint with double-cover plates only gives 79 per cent. of the plate strength. This is a very serious loss of strength, and various attempts have been made to bring up the strength of the joint to that of the solid plate. With this object in view, Sir Wm. Fairbairn patented a process of rolling plates with thickened edges, so that, after the holes were punched and the plates riveted, the sectional area of the plate between the rivets, would be equal to that of the solid plate. This plan, however, has never been adopted in practice, probably on the ground of expense.

A proposal has also been made to make the rivets of an elliptical section, so that by keeping the same pitch for the same sectional area of rivets there would be greater breadth of plate between the rivet holes, than when round rivets are used. Thus, if instead of using round rivets 1 inch diameter, rivets of equal area with flat sides (say $1\frac{1}{4}$ inch × $\frac{3}{4}$ inch) are used, and are fixed in position with the flat sides parallel, there is a gain in the

breadth of the plate between the rivet holes of $\frac{1}{4}$ inch. There are many objections, however, to this form of rivet, and it has not yet come into use.

Caulking.—In order that the riveted joints of boilers may be absolutely steam and water tight, they usually require to be caulked. This consists in burring down the edges of the plates with a tool somewhat like a chisel, but flat on the end (see fig. 2). Caulking, whilst indenting down the extreme edge of the lap, is liable to open the plates between the extreme edge and the point where they are held by the rivet-heads. For this reason, many engineers have given up the use of the caulking tool, and prefer to use only the fullering tool.—See fig 1. Caulking or fullering is greatly facilitated if the plates are planed on the edges with a slight bevel, and that is now done in the best boiler practice. The caulking tool recommended by Mr. Webb, Locomotive Superintendent of the London & North-Western Railway, and used in the works of that Railway Company at Crewe, is shown in the accompanying diagram, by which damage to the plates is prevented and the surfaces driven into close contact.

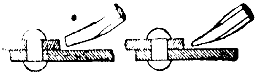
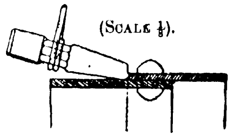


Fig. 1. (SCALE $\frac{1}{8}$) Fig. 2.



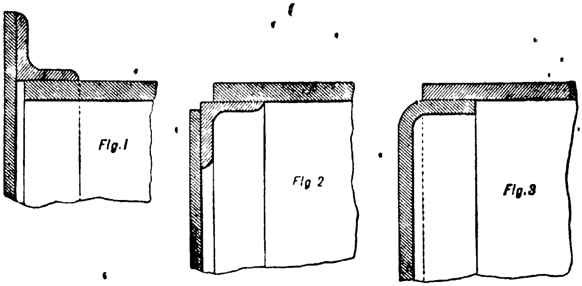
(SCALE $\frac{1}{8}$).

WEBB'S CAULKING TOOL.

Welded Joints.—In recent years welded joints have been introduced for boiler shells, but have not met with much favour. All chances of leakage are avoided by welding the joints, and the external corrosion which results from leakage is therefore prevented. If the joint is sound, it is stronger than any of the forms of riveted joint, but its soundness is always a matter of uncertainty. The strength of a riveted joint can be known with a considerable degree of accuracy, but the strength of a welded joint depends entirely on the skill and care of the workmen, and it is not always easy to decide from the external appearance whether or not the weld is sound. Welded joints, however, are very serviceable for furnace tubes and locomotive steam domes, &c., and are in general use for those purposes, but they have not been adopted to any great extent for boiler shells.

Methods of connecting the Parts of the Shell and Flues.—The boiler shell is composed of rings of plating from three to four feet six inches in width, rolled with the grain running circumferentially. These rings are connected to each other at their edges by lap or butt joints. The flat end plates are connected to

the shell in various ways; the following diagram shows the three methods that are most common in practice.

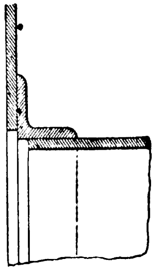


Figs. 1 and 2 show the method of attachment by riveting angle irons to the shell and then riveting the end plate to those angle irons. In fig. 1, the angle iron is attached *outside*, and in fig. 2, *inside* the shell. These two methods are largely used in land boilers, notably Lancashire and Cornish boilers. The attachment by outside angle irons admits of more springing of the end plates, and gives more room for mountings, &c., on the front end of the boiler. The outside angle irons are preferable when the space between the flue tubes and the shell plates does not exceed 5 or 6 inches, owing to the greater freedom allowed by them for the longitudinal expansion of the flue. A very common arrangement in land boilers is to attach the front end plate by outside angle irons, and the back end plate by inside angle irons.

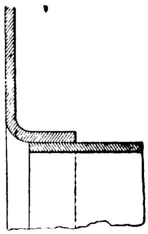
Fig. 3 shows the method of flanging the end plate and then riveting it to the shell plates by an ordinary lap joint. This is the form most generally used in marine practice, and now also to a large extent for land boilers. It forms the best and simplest form of joint, but of course the end plate must be of thoroughly good quality in order to stand the flanging. When the end plate is attached by angle irons to the shell, the constant springing which goes on, due to the expansion and contraction of the furnace flues, is liable to cause grooving of the end plates close to the edge of the angle iron: but when the end plates are flanged, the bending is spread over the curvature at the root of the flange, and is not concentrated upon any particular point, consequently grooving is prevented to a great extent.

There are other methods of attaching the flat end plates to the

shell, such as flanging the end plates outwards, instead of inwards as shown, or fitting angle irons outside instead of inside the end plate, but these are not in such general use, and the only advantage they possess is, that the joint may be riveted wholly by machinery.

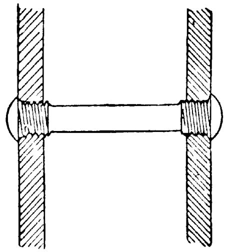


Flue Attachment.—The methods of attaching the flues to the end plates are very similar to those already described for fixing the shell to the end plates. Two of those in common use are shown in the diagram at the side. In one arrangement, an angle iron is used, and in the other, the end plate is flanged inwards. It is also sometimes flanged outwards.



Staying of Boilers.—The tendency of pressure on a flat surface is to bulge it out to a circular form, and to prevent this deformation of the flat surfaces in steam boilers, all such surfaces require to be stayed. In Lancashire and Cornish boilers the only parts which require staying are the end plates; in marine boilers the end plates and the flat sides of the combustion chambers, and in locomotive boilers all the flat sides of the fire-box and the end plates.

Stays are made of various forms according to the position they occupy in the boiler. The fire-box stays of locomotives, which bind the flat sides of the fire-box to the outer shell, are shown in the annexed diagram.

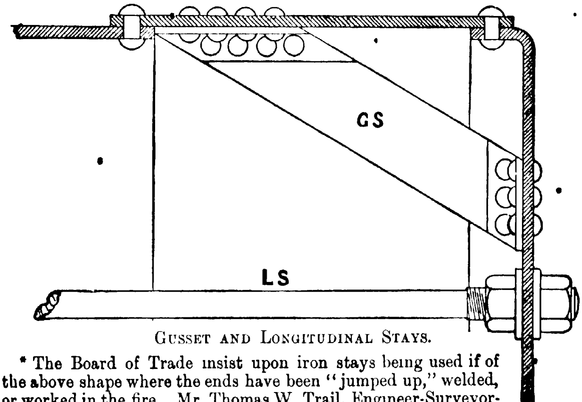


SCREW STAYS.

The stays of marine boilers which bind the flat sides of the combustion chambers together, and to the end plates are similar in form, but have often nuts and washers on one or on both ends instead of riveted heads, and are generally screwed throughout their length. In locomotive boilers these stays are usually made of copper, but in marine boilers wrought iron or steel is used. Nuts and washers on the ends of the stays give better support to the plates than riveted heads, owing to their larger bearing surface.

The end plates of boilers are stayed with gusset stays, or with longitudinal stays passing from end to end of the boiler, or with both.

Gusset stays are usually made of a single plate of iron, which is fixed to the shell and to the end plate, by means of angle irons on each side of the plate, as shown at, G S, on the following diagram. In marine boilers, the gusset stays are often made of a rod of iron with a flat plate forged on the end. The plate is riveted to the shell, and the rod passes diagonally across to the end plate, and is fixed there by nuts and washers on each side.



GUSSET AND LONGITUDINAL STAYS.

* The Board of Trade insist upon iron stays being used if of the above shape where the ends have been "jumped up," welded, or worked in the fire. Mr. Thomas W. Trail, Engineer-Surveyor-in-Chief to the Board of Trade, in his book on Boilers, published by Chas. Griffin & Co., gives the pressures, greatest surfaces, and sizes for iron and steel stays. See pp. 126 to 137 and 258 to 263, fifth edition.

Longitudinal stays are simply rods of iron or steel* screwed at the ends, which pass from one end of the boiler to the other, and are secured to the end plates by nuts and washers on both sides. One of these, L S, is shown on the above diagram. When these stays exceed 20 feet or thereby in length they tend to droop in the centre, and do not take up the full stress on the end plates. In this case, they should be supported at the centre by small brackets riveted to the shell.

Although the end plates require this staying, it is not desirable that they should be absolutely rigid, or the flues will not have sufficient freedom to expand. The object to be aimed at is to strengthen the ends, but yet as far as possible to preserve a certain amount of elasticity

The flat crowns of locomotive fire-boxes, and the combustion chambers of marine boilers, are usually stayed in the manner shown on the diagram of multitubular boiler in Lecture XIX.,

Vol. I., and the marine boiler, p. 334, in Lecture XVIII. Cast-iron or preferably wrought-iron girder plates pass across the fire-box and rest on the vertical back and front plates. These girder plates support the furnace crown by means of bolts passing up through them, and are secured by nuts above, in the manner shown on the diagrams. In marine boilers, the combustion chamber is sometimes curved at the top and supported by stays from the end plate (p. 331), but it is usually flat on the top and supported in the same way as the locomotive fire-box just noticed. The top of a locomotive fire-box is sometimes also supported by copper or iron stays from the outer shell of the boiler, in the same way as the sides (see folding page, Lecture XXVI., Vol. I.). In regard to crown stays, Mr. D. S. Smart, in his paper on "Steam Boilers," read before the Institution of Civil Engineers,* remarks—

"Girder-stays have, until recently, been universally employed in the strengthening of fire-box crowns of the locomotive type of boilers; but direct stays between the crowns of the fire-box casings and the fire-box crowns are now to a great extent taking their place. Girder stays are decidedly objectionable in obstructing the circulation of the water, and in tending to cause overheating through the narrow water spaces between them and the crowns becoming choked with deposit; also on account of the severe stress thrown upon the plates on which they rest. The object in refraining from staying the crowns of the fire-box directly to the crowns of the casings has hitherto been to avoid undue strain from the greater upward expansion of the fire-box, but this objection may in a great measure be overcome, by making the crowns of the casings flat like the fire-box crowns with well rounded corners. The pressures on the two flat surfaces will nearly balance, and any unequal expansion will be taken up by the flat portions outside the stays, or by the rounded corners. The Author has seen a number of boilers constructed on this design which he believes will give perfect satisfaction. The two crowns are stayed directly to each other by bolts screwed into both, with the heads in the fire-box and nuts on the top of the outer casing, the part in the water and steam spaces being without threads. Numbers of boilers are also being made with the crowns of the casings of the usual semi-cylindrical form, and the flat fire-box crowns stayed directly to them by bolts in the manner just described, with no provision for expansion other than the spring of the plates all round. Others, when thus arranged, especially when the fire-boxes are of large size, have provision for the upward expansion of the first two rows of stays

* Volume lxxx. of Proceedings. Extract and diagrams taken from it by kind permission of the Council of the Inst. of C.E.

over the tube plate on getting up steam, as tube plates have been injured by too rigid a connection."

In marine and all tubular boilers, that part of the end plate through which the tubes pass, and which cannot be supported by gusset stays, is supported by some of the tubes themselves which are known as *stay* tubes. These stay tubes are made stronger than the others, and are usually screwed into the back and the front tube plates. They are sometimes fitted with nuts on the outside of the front tube plate and then beaded over at each end. The tube plates are seldom supported by rod stays between the tubes, for this plan is objectionable, since the rods are not exposed to the same temperature as the tubes, and consequently expand differently.

Strength of Boiler Shells.—The strongest form for any boiler, or vessel which supports internal pressure, is that of a sphere, but there are many reasons for not adopting this form in practice. The early steam boilers were designed of the form which would give most heating surface, and in the opinion of the designer would give the highest evaporative efficiency, but no attention was given to the form which would be best adapted to support pressure. So long as very low pressure steam was used in those boilers, the question of form was not so important, but as soon as pressures of 30 lbs. per square inch or thereby were adopted, it became necessary to give some attention to the form of the shell which is best suited to withstand internal pressure. The cylindrical boiler has now been universally adopted as the nearest practical approach to the sphere.

To estimate the strength of a cylindrical boiler—

Let P = the bursting pressure in lbs. per square inch.
„ t = thickness of the plates in inches.
„ D = diameter of the boiler in „
Let S_t = tensile strength of the material at its weakest part in lbs. per square inch.

Consider the pressure on any very small surface, A B, (in the next diagram) which makes an angle, θ, with the horizontal diameter, O D. The normal pressure, P, on the surface, A B, may be resolved into two components, Ox, acting vertically, and, Oy, acting horizontally.

$$\text{Then the angle } EOy = \theta,$$
$$\therefore \quad \text{„} \quad \text{„} \quad POy = 90 - \theta,$$
$$\text{Therefore } x = P \cdot \sin. (90 - \theta),$$
$$= P \cdot \cos. \theta.$$

Thus the vertical pressure on the surface, $AB = P\cos.\theta \times AB$.

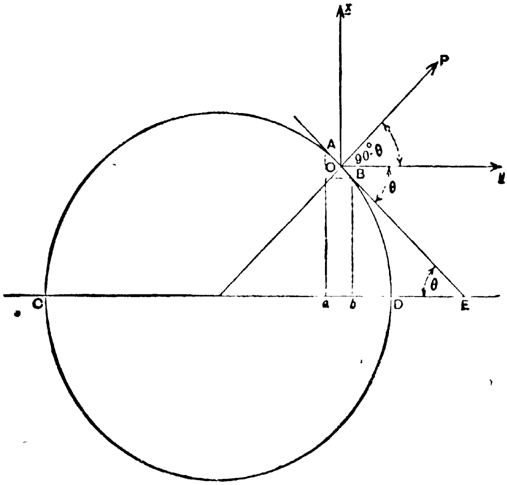
$$\text{But } \cos.\theta \cdot AB = ab.$$

$$\therefore \text{ The vertical pressure on } AB = P \times ab.$$

Hence, the sum of all the vertical components of, P, will be—

$$P \times OD = P \times D.$$

i.e., The force tending to rupture the boiler is equal to the pressure per square inch, multiplied by the diameter in inches.



Also, the resistance of the material is equal to the tensile strength of the plates in lbs. per square inch, multiplied by the combined area of the plates on each side of a diameter.

Then at the point of rupture—

The pressure tending to cause rupture = The resistance of the material.

Considering the pressure on any length, L, of the boiler,

We have—

$$P \times D \times L = 2(L \times t \times S_t)$$

$$\therefore P = \frac{2\,t\,S_t}{D}$$

$$\text{and } t = \frac{P\,D}{2\,S_t}$$

The value of P, given by the previous formula, is the pressure required to cause *longitudinal rupture, i.e.*, rupture in a line parallel to the axis of the boiler, but a cylindrical boiler may also be ruptured transversely, *i.e.*, in a line at right angles to the axis, due to the pressure on the ends.

Let P_1 = the bursting pressure in this case.

Then the force tending to cause rupture—

$$= P_1 \times \text{area of cross section of boiler} = P_1 \times \frac{\pi}{4} D^2.$$

Also, resistance of plates = area of metal in cross section × its tensile strength.

i.e., Force tending to cause rupture = Resistance of plates.

$$P_1 \times \frac{\pi}{4} D^2 = \pi D \times t \times S_t$$

$$\therefore \; P_1 = \frac{\pi D t S_t}{\frac{\pi}{4} D^2} = \frac{4 t S_t}{D}$$

$$\therefore \; \frac{P}{P_1} = \frac{\frac{2 t S_t}{D}}{\frac{4 t S_t}{D}} = \frac{1}{2}.$$

Hence, the pressure required to cause rupture of a boiler longitudinally, is only half that required to cause rupture in a transverse direction. For this reason, the longitudinal joints of boilers are always made stronger than the circumferential joints. In Cornish, Lancashire, and marine boilers having internal flues from end to end, the pressure required to cause transverse rupture is much greater than twice that required to cause longitudinal rupture, for then the effective area of the end plates is not equal to the whole area of cross section of the boiler, but is equal to the area of the boiler minus the area of the flues. Owing to this unequal stress on the joints of boilers, it has been proposed to plate boilers diagonally, having the joints at such an angle to the axis as would cause an equal stress on each joint. This plan, however, has never yet been put into actual practice.

In all actual calculations we must insert for, S_t, not the tensile strength of the plate, but the strength of the riveted joint. This is obtained by taking the percentage of plate strength given in the table on page 373, for the particular form of riveted joint with which the boiler is constructed.

EXAMPLE.—A Lancashire boiler is 7 feet 6 inches diameter, and

is required to work at a pressure of 75 lbs. per square inch. The longitudinal joints are double-riveted and are lap joints.

Find the thickness of wrought-iron plates required for the shell.

The average tensile strength of wrought-iron plates in the direction of the grain is 21 tons, or 47,040 lbs. per square inch (see page 364), and since a double-riveted lap joint (punched holes) gives 69 per cent. of plate strength,

The tensile strength of joint = 47,040 × ·69.
" " " = 32,457 lbs. per square inch.

In steam boilers, a factor of safety of 6, is usually allowed, *i.e.*, the bursting pressure is six times the working pressure.

∴ The bursting pressure = 75 × 6.
" " " = 450 lbs. per square inch.

$$\text{Then,} \quad t = \frac{P\,D}{2\,S_t} = \frac{450 \times 90}{2 \times 32457}$$

$$\therefore t = \cdot 6239'' = \frac{5}{8} \text{ inch nearly.}$$

Strength of Flues.—The strength of cylindrical tubes subjected to *internal* pressure is independent of the length of the tube, since the greater the length of the tube the more material there is to resist the increased pressure. In proof of this statement the student will have noticed that in the equation on page 382, the quantity, L, appeared on both sides, and was therefore cancelled out. It seems natural also to suppose that when a cylindrical tube is subjected to *external* pressure, its strength to resist collapse should not be dependent upon its length, and until the year 1858, this was assumed by all engineers.

Sir Wm. Fairbairn carried out a series of experiments in 1858, to ascertain the strength of cylindrical tubes subjected to external pressure, and his experiments threw much light on the behaviour of such tubes under those conditions.

As the result of his experiments, he deduced the following formula for the strength of boiler flues of iron :—

Let P = collapsing pressure in lbs. per square inch.
" t = thickness of the plates of the flue in inches.
" L = length of the flue in feet.
" D = diameter of the flue in inches.

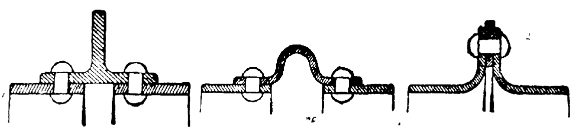
$$\text{Then, } P = 806{,}300\,\frac{t^{2.19}}{L \times D}$$

This shows that the collapsing pressure varies directly as the 2·19th power of the thickness, and inversely as the length and diameter.

When a cylindrical tube, which is not perfect in form, is subjected to *internal* pressure, the effect of the internal pressure is to rectify the defect, and to bring the tube to the form of an exact cylinder. Thus, in a boiler shell made with lap joints, the form of the cross section necessarily differs from that of a true circle, but when steam pressure comes upon it, the tension on the longitudinal joints tends to draw the plates into line, and causes them to take up an exactly circular form at any section. The effect of *external* pressure on an imperfectly circular tube is not to remedy the imperfection, but to increase the deviation from the true circular section and to produce greater distortion. Among Sir Wm. Fairbairn's experiments, the results may be noted of a test of two tubes subjected to external pressure, 37 in. long, 9 in. diameter, and ·14 in thick, the same in every respect, except that one tube was lap jointed and the other butt jointed. The tube with the lap joint collapsed with 262 lbs. pressure per square inch, whilst the tube with the butt joint did not give way till a pressure of 378 lbs. per square inch was reached. This shows a loss of $\frac{1}{3}$ in resistance to collapse, by a departure of merely ·14 in. from the true circular section, and clearly indicates the necessity of making boiler flues *exactly cylindrical.* This is now very nearly approached in practice, flues being always made with either welded or butt joints.

Fairbairn's formula may be readily worked out by the use of logarithms; but for ordinary practical purposes, the *square* of the thickness may be used instead of the 2·19th power.

From the above, it will be apparent that cylindrical tubes, subjected to external pressure, require to be strengthened when long. Boiler flues are usually strengthened at intervals along their length, and this is effected in several ways, the principal of which are—



Scale $\frac{1}{8}$

T IRON RING. BOWLING HOOP. ADAMSON'S FLANGED SEAM.

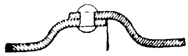
The T iron ring shown in section in the left-hand figure was

the first method adopted for strengthening the flues. It is riveted round the joints of each ring of plates in the manner shown in the diagram. This plan gives ample strength, but holds the flue too rigidly, and does not permit of free expansion and contraction. The rivet heads also are exposed to the intense heat of the furnace, and are liable to be burnt.

The form shown in the middle figure is known as the Bowling-hoop, and has been largely used for strengthening the furnace flues of boilers. It is weldless, and can be made in iron or steel. It possesses quite as great strength as the T iron ring, and, from its shape, allows all necessary freedom for the expansion of the flue. It has the same disadvantage as the T iron ring, however, since it exposes a double thickness of plates and two rows of rivets to the flames from the fire-grate.

The third method shown in the right hand figure is known as the Adamson Flanged Seam, and consists in flanging the ends of the flue plates, and connecting them together by rivets with a ring between. This joint is very elastic, and permits of free expansion; it has sufficient strength without the ring, but the ring is used in order to give a caulking edge on each side of the lap. This is the method which is most generally adopted, although a number of engineers prefer the Bowling-hoop. The plates require to be of specially good quality to admit of flanging, and the flanging must be skilfully done, or the joint will give a considerable amount of trouble. The advantage of this joint is, that all rivets and double thicknesses of plate are removed from the action of the fire.

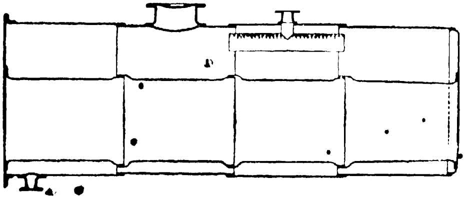
There is one other method of strengthening flues which is of more recent introduction than those already mentioned, and is shown in the diagram annexed.



PAXMAN'S FLUE JOINT

The flue tubes are made of welded rings of iron or steel, and are rolled out accurately in a machine to the shape shown, and united by a simple lap joint. The strength of the flues is thus greatly increased, and yet free expansion is allowed

The arrangement of these flue tubes in a boiler is shown in the diagram below.



The rivet heads and double thicknesses of plate, although not removed entirely from the action of the flames, are out of immediate contact with them.

When flues are strengthened by any of these methods, their strength must be taken as that corresponding to the length between the rings or joints.

Corrugated Furnaces.—The furnaces of boilers are now very frequently fitted with corrugated flues. A furnace of this kind is shown in the marine boilers illustrated on pages 335 and 337. A corrugated furnace flue is stronger than one fitted with any of the strengthening rings already mentioned, and is of such a form as to allow every facility for expansion. The process of corrugating furnace flues was brought out and patented by Mr. Samson Fox in 1876. The appliances at first used for producing the corrugations were very severe and trying to the material, but now by the use of improved rolling mills, corrugated furnaces are produced in which the plates suffer no apparent injury.

Within the last few years corrugated furnace flues have been largely used, both for land and for marine boilers, with very satisfactory results. Greatly increased strength, combined with perfect elasticity, is the principal advantage, but a corrugated furnace also gives greater heating surface, and breaks up the flame and heated gases. They have, however, certain disadvantages, viz., sediment and salt incrustation may more readily gather in the hollows at the top of the flue than in a plain cylindrical one, and the dead ashes in the lower inside hollows. The elasticity or bellows action is somewhat too great in large boilers, and strengthening longitudinal stays are sometimes inserted round or near the outside of the Fox's tubes between the ends of the boiler.

LECTURE XX.—QUESTIONS.

1. Enumerate the chief advantages of wrought-iron as a suitable material for the construction of steam boilers. What kinds of iron plates should be discarded, and why? Give the average tensile strength of good wrought-iron with and across the grain.

2. State the chief advantages of mild steel over wrought-iron as a material for boiler construction, and explain the precautions that are necessary in selecting the plates and in manipulating them during the manufacture of a boiler. Give any instances known to you of the failure of steel boiler plates, and the reasons assigned for their failure. Give the tensile strength of good mild boiler plate steel.

3. In what kinds of boilers, and for what parts of them, is copper used? What advantages are claimed for copper in those cases over wrought-iron or steel? Why is cast-iron not used for the shell or flues of ordinary land or marine boilers? In what kinds of boilers, and for what parts of them, is cast-iron still employed?

4. Sketch clearly in freehand the several chief forms of riveted joints, and indicate the advantages and disadvantages of each. Sketch a single-riveted joint for $\frac{3}{8}$-inch plate, marking the size of the rivets and the pitch you would employ. Show in what way such a joint might yield.

5. Determine the pitch of the rivets for a single riveted joint of $\frac{1}{4}$-inch plate so that the joint may be equally strong to resist tearing and shearing. Diameter of rivets is $\frac{3}{4}$-inch. Safe shearing strength is 7,800 lbs. per square inch; safe tensile strength is 10,000 lbs. per square inch. *Ans.* 1·813 inch.

6. What rules are employed for calculating the strength of double-riveted lap joints in iron and steel plates? Is there any advantage in the use of elliptical rivets?

7. It is required to construct a double-riveted lap joint for $\frac{1}{2}$-inch plates. Give the proportions of the joint, and calculate the percentage of solid plate strength which it gives.

8. Describe with sketches any boiler or other piece of riveted work with which you have had anything to do. Give roughly the dimensions of rivets or stays, the details of joints; give fuller information about the part you have had most to do with. What sort of stress occurs in the plates at any riveted joint? Sketch the various ways in which fracture may occur.

9. What are the relative advantages and disadvantages of different methods of riveting?

10. How are rivets made, and from what kinds of iron? Sketch, with dimensions, single- and double-riveted butt joints of $\frac{5}{8}$-inch plates. Show a butt joint in a boiler where cross and longitudinal joints meet. Sketch the various ways in which the joint may be made at the bottom of a locomotive fire-box.

11. In a single-riveted lap joint exposed to tension, determine the diameter and pitch of the rivets in terms of the thickness of plate, and the three stresses f_t, f_s, and f_b,

where f_t = intensity of stress on material of plate.

f_s = " " " rivet.

f_b = " bearing pressure estimated on a diametral section of rivet.

Find the diameter (d) and pitch (p) for $\frac{1}{2}$-inch plates when $f_t = 30$, $f_s = 22$, and $f_b = 42$ tons per square inch, and estimate the efficiency of the joint. *Ans.* $p = 2·88$ inches, and $d'' = 1·21$ inch; 50 per cent.

12. Compare the joints of plates in the end of a Cornish or Lancashire boiler now with what they were twenty years ago.

13. Which is better, to drill or punch rivet holes? and why?

14. State the chief objections to punching boiler plates. Why should "drifting" the holes not be permitted in order to bring them fairly opposite each other? What is the best method for ensuring that the rivet holes in boiler plates shall be fairly opposite each other?

15. Why is hydraulic machine-riveting better than hand-riveting?

16. Describe the process of caulking a joint, and sketch the best form of caulking-tool with which you are best acquainted.

17. In what parts of boilers are welded joints used? Why are welded joints not more generally adopted?

18. Sketch and describe the principal plans of connecting the shell to the end plates in a large horizontal boiler, and give their several advantages and disadvantages, with reasons.

19. Sketch and describe the best plans of connecting the flues to the end plates of a horizontal land or marine boiler.

20. Mention those parts of a marine boiler which require to be stayed, and show clearly by sketches how the staying is done in actual practice.

21. Find the thickness of iron plates in a boiler shell 6 feet 4 inches in diameter, for a pressure of 40 lbs., the greatest tensile stress permissible in the material being 5,000 lbs. per square inch. *Ans.* ·304 inch.

22. A cylindrical boiler with flat ends, 30 feet long, 6 feet diameter, has two internal flues, each $2\frac{1}{4}$ feet in diameter. Steam pressure in the boiler is 40 lbs.; what is the whole pressure on the internal surface in tons? How is the strength of such a boiler related to its diameter? *Ans.* 10·8 tons.

23. Find the greatest diameter of a cylindrical boiler to resist a pressure of 100 lbs. per square inch, the plates being $\frac{3}{8}$-inch thick, and the safe stress upon the metal being 5,500 lbs per square inch. *Ans.* 41·25 inches.

24. A cylinder constructed of boiler plate is 7 feet in diameter, and is subjected to an internal bursting pressure of 50 lbs. per square inch. Find the longitudinal stress on the metal per square inch of section, the thickness of the plate being $\frac{1}{2}$ inch. *Ans.* 4,200 lbs.

25. Show fully by calculation why a cylindrical boiler is twice as likely to burst longitudinally as endwise, and give an example.

26. An ordinary cylindrical boiler has flat ends with two internal flues running from end to end. The boiler is 28 feet long, the shell 7 feet in diameter, and each of the two flues is 30 inches in diameter, the iron employed being $\frac{1}{2}$-inch in thickness throughout. Taking the ultimate strength for the longitudinal or double-riveted joints at 35,000 lbs. per square inch of sectional area, and that for the transverse or single-riveted joints at 28,000 lbs. per square inch, find the ultimate bursting pressure—(1) along a longitudinal, (2) along a transverse section. *Ans.* 417; 1,532 lbs

In what way are the internal flues strengthened?

27. Given the breaking tensile strength of wrought-iron, find the thickness of the shell of a cylindrical boiler which will support a given pressure of steam. Example—The diameter of the shell is $3\frac{1}{2}$ feet, and the pressure of the steam is 150 lbs. on the square inch, what should be the thickness of the boiler plate when the tensile strength of wrought-iron is, *for safety*, estimated at three tons on the square inch? Prove that a tube under internal fluid pressure is twice as strong in a transverse as in a longitudinal direction. *Ans.* ·47 inch.

28. The furnace flue of a marine boiler is 7 ft long and 3 ft. in diameter The plates are $\frac{3}{8}$-inch thick, and an Adamson flanged joint is fitted at the centre. Find the collapsing pressure of the flue. *Ans.* 374 lbs. per sq. inch

29. Find an expression for the thickness of the shell of a cylindrical boiler, the tensile strength of the material, the pressure of steam, and the diameter of the shell being given. If a cylindrical boiler 5 feet in diameter will support a steam pressure of 20 lbs., what should be the diameter of a boiler of like material, construction, and thickness of plate, for supporting a steam pressure of 100 lbs.?

30. In a cylindrical steam boiler prove the formulæ for the forces tending to produce rupture of the material in the circumferential and longitudinal directions.

31. Why do ordinary steam boilers fail to utilise a large proportion of the heat developed in the complete combustion of the fuel employed? Sketch a longitudinal section through the fire-box and tubes of a high-pressure boiler. Describe the construction of the boiler, and show the method of staying the portions most likely to give way under pressure

32. Answer only one of the following questions (*a*, *b*, or *c*):—(*a*) Describe, with the aid of sketches, the boiler with which you are best acquainted, particular attention to be paid to the fittings, to the flues, and to the furnace. (*b*) Sketch and describe the construction of the locomotive fire-box; show in detail how the flat sides are supported, and indicate the form and construction of the stay-bolts. If the boiler pressure be 175 lbs. absolute and the stays are placed 4 inches from centre to centre, find the tensile force in each stay-bolt and the diameter which the bolts or stays must have at their weakest point in order that the metal may not be stressed beyond 5 tons to the square inch of their section. (*c*) Sketch and describe fully the construction of a cylindrical or piston slide-valve and the cylinder ports for the same. How is the ring prevented from springing into the port opening?

33. Design a double-riveted lap joint for a boiler, the plating being $\frac{3}{4}$ of an inch thick and of steel. Choose your own strength constants, and take account of the bearing strength of the material as well as the liability to tear and shear.

34. A cylinder 30 feet long and 4 feet in diameter has to be designed to resist an internal fluid pressure of 300 lbs per square inch above the atmospheric pressure. Choosing for yourself the various working stresses to be allowed, determine (*a*) the thickness of the steel plate you would use, (*b*) the diameter of rivets and their pitch in the double-riveted butt joints for the longitudinal seams

35. A Lancashire boiler, 30 feet long and 7 feet 6 inches diameter, is required to work at a pressure of 100 lbs per square inch, the material of the shell being steel. Design a suitable form of longitudinal joint, choosing your own working stresses, &c

APPENDIX.

EXAMINATION TABLES.

USEFUL CONSTANTS.

1 Inch = 25·4 millimetres.

1 Gallon = ·1605 cubic foot = 10 lbs. of water at 62° F. ∴ 1 lb. = ·01605 cubic foot.

1 Knot = 6080 feet per hour. 1 Naut = 6080 feet.

Weight of 1 lb. in London = 445,000 dynes.

One pound avoirdupois = 7000 grains = 453·6 grammes.

1 Cubic foot of water weighs 62·3 lbs.

1 Cubic foot of air at 0° C. and 1 atmosphere, weighs ·0807 lb.

1 Cubic foot of Hydrogen at 0° C and 1 atmosphere, weighs ·00557 lb.

1 Foot-pound = $1·3562 \times 10^7$ ergs.

1 Horse-power-hour = 33000 × 60 foot-pounds.

1 Electrical unit = 1000 watt-hours.

Joule's Equivalent to suit Regnault's H, is $\begin{cases} 774 \text{ ft.-lbs} = 1 \text{ Fah. unit.} \\ 1393 \text{ ft. lbs.} = 1 \text{ Cent. } ,, \end{cases}$

1 Horse-power = 33000 foot-pounds per minute = 746 watts.

Volts × amperes = watts.

1 Atmosphere = 14·7 lb. per square inch = 2116 lbs. per square foot = 760 m.m. of mercury = 10^6 dynes per sq. cm. nearly.

A Column of water 2·3 feet high corresponds to a pressure of 1 lb. per square inch.

Absolute temp., t = θ° C. + 273°·7.

Regnault's H = 606·5 + ·305 θ° C = 1082 + ·305 θ° F.

$p\,u^{1.0646} = 479$

$\log_{10} p = 6{\cdot}1007 - \frac{B}{t} - \frac{C}{t^2}$,

where $\log_{10} B = 3{\cdot}1812$, $\log_{10} C = 5{\cdot}0871$,

p is in pounds per square inch, t is absolute temperature Centigrade, u is the volume in cubic feet per pound of steam.

$\pi = 3{\cdot}141\dot{6} = \frac{22}{7} = \frac{355}{113} = 10(\sqrt{3} - \sqrt{2})$.

One radian = 57·3 degrees.

To convert common into Napierian logarithms, multiply by 2·3026.

The base of the Napierian logarithm is e = 2·7183.

The value of g at London = 32·182 feet per second per second.

TABLE OF LOGARITHMS.

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
10	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	4	8	12	17	21	25	29	33	37
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	4	8	11	15	19	23	26	30	34
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	3	7	10	14	17	21	24	28	31
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	3	6	10	13	16	19	23	26	29
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	3	6	9	12	15	18	21	24	27
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	3	6	8	11	14	17	20	22	25
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	3	5	8	11	13	16	18	21	24
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	2	5	7	10	12	15	17	20	22
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	2	5	7	9	12	14	16	19	21
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	2	4	7	9	11	13	16	18	20
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	2	4	6	8	11	13	15	17	19
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	2	4	6	8	10	12	14	16	18
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	2	4	6	8	10	12	14	15	17
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	2	4	6	7	9	11	13	15	17
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	2	4	5	7	9	11	12	14	16
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	2	3	5	7	9	10	12	14	15
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	2	3	5	7	8	10	11	13	15
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	2	3	5	6	8	9	11	13	14
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	2	3	5	6	8	9	11	12	14
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	1	3	4	6	7	9	10	12	13
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	1	3	4	6	7	9	10	11	13
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	1	3	4	6	7	8	10	11	12
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	1	3	4	5	7	8	9	11	12
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	1	3	4	5	6	8	9	10	12
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	1	3	4	5	6	8	9	10	11
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	1	2	4	5	6	7	9	10	11
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	1	2	4	5	6	7	8	10	11
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	1	2	3	5	6	7	8	9	10
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	1	2	3	5	6	7	8	9	10
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	1	2	3	4	5	7	8	9	10
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	1	2	3	4	5	6	8	9	10
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	1	2	3	4	5	6	7	8	9
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	1	2	3	4	5	6	7	8	9
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	1	2	3	4	5	6	7	8	9
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	1	2	3	4	5	6	7	8	9
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	1	2	3	4	5	6	7	8	9
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	1	2	3	4	5	6	7	7	8
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	1	2	3	4	5	5	6	7	8
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	1	2	3	4	4	5	6	7	8
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	1	2	3	4	4	5	6	7	8
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	1	2	3	3	4	5	6	7	8
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	1	2	3	3	4	5	6	7	8
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	1	2	2	3	4	5	6	7	7
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	1	2	2	3	4	5	6	6	7
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	1	2	2	3	4	5	6	6	7

TABLE OF LOGARITHMS—*Continued.*

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	1	2	2	3	4	5	5	6	7
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	1	2	2	3	4	5	5	6	7
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	1	2	2	3	4	5	5	6	7
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	1	1	2	3	4	4	5	6	7
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	1	1	2	3	4	4	5	6	7
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	1	1	2	3	4	4	5	6	6
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	1	1	2	3	4	4	5	6	6
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	1	1	2	3	3	4	5	6	6
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	1	1	2	3	3	4	5	5	6
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	1	1	2	3	3	4	5	5	6
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189	1	1	2	3	3	4	5	5	6
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	1	1	2	3	3	4	5	5	6
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	1	1	2	3	3	4	5	5	6
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	1	1	2	3	3	4	4	5	6
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	1	1	2	2	3	4	4	5	6
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	1	1	2	2	3	4	4	5	6
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	1	1	2	2	3	4	4	5	5
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	1	1	2	2	3	4	4	5	5
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686	1	1	2	2	3	4	4	5	5
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	1	1	2	2	3	4	4	5	5
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	1	1	2	2	3	3	4	5	5
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	1	1	2	2	3	3	4	5	5
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	1	1	2	2	3	3	4	4	5
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	1	1	2	2	3	3	4	4	5
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	1	1	2	2	3	3	4	4	5
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	1	1	2	2	3	3	4	4	5
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	1	1	2	2	3	3	4	4	5
82	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186	1	1	2	2	3	3	4	4	5
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	1	1	2	2	3	3	4	4	5
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	1	1	2	2	3	3	4	4	5
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	1	1	2	2	3	3	4	4	5
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390	1	1	2	2	3	3	4	4	5
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	0	1	1	2	2	3	3	4	4
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	0	1	1	2	2	3	3	4	4
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	0	1	1	2	2	3	3	4	4
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	0	1	1	2	2	3	3	4	4
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	0	1	1	2	2	3	3	4	4
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	0	1	1	2	2	3	3	4	4
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	0	1	1	2	2	3	3	4	4
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	0	1	1	2	2	3	3	4	4
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	0	1	1	2	2	3	3	4	4
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	0	1	1	2	2	3	3	4	4
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	0	1	1	2	2	3	3	4	4
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	0	1	1	2	2	3	3	4	4
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	0	1	1	2	2	3	3	3	4

TABLE OF ANTILOGARITHMS.

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
·00	1000	1002	1005	1007	1009	1012	1014	1016	1019	1021	0	0	1	1	1	1	2	2	2
·01	1023	1026	1028	1030	1033	1035	1038	1040	1042	1045	0	0	1	1	1	1	2	2	2
·02	1047	1050	1052	1054	1057	1059	1062	1064	1067	1069	0	0	1	1	1	1	2	2	2
·03	1072	1074	1076	1079	1081	1084	1086	1089	1091	1094	0	0	1	1	1	1	2	2	2
·04	1096	1099	1102	1104	1107	1109	1112	1114	1117	1119	0	1	1	1	1	2	2	2	2
·05	1122	1125	1127	1130	1132	1135	1138	1140	1143	1146	0	1	1	1	1	2	2	2	2
·06	1148	1151	1153	1156	1159	1161	1164	1167	1169	1172	0	1	1	1	1	2	2	2	2
·07	1175	1178	1180	1183	1186	1189	1191	1194	1197	1199	0	1	1	1	1	2	2	2	2
·08	1202	1205	1208	1211	1213	1216	1219	1222	1225	1227	0	1	1	1	1	2	2	2	3
·09	1230	1233	1236	1239	1242	1245	1247	1250	1253	1256	0	1	1	1	1	2	2	2	3
·10	1259	1262	1265	1268	1271	1274	1276	1279	1282	1285	0	1	1	1	1	2	2	2	3
·11	1288	1291	1294	1297	1300	1303	1306	1309	1312	1315	0	1	1	1	2	2	2	2	3
·12	1318	1321	1324	1327	1330	1334	1337	1340	1343	1346	0	1	1	1	2	2	2	2	3
·13	1349	1352	1355	1358	1361	1365	1368	1371	1374	1377	0	1	1	1	2	2	2	3	3
·14	1380	1384	1387	1390	1393	1396	1400	1403	1406	1409	0	1	1	1	2	2	2	3	3
15	1413	1416	1419	1422	1426	1429	1432	1435	1439	1442	0	1	1	1	2	2	2	3	3
·16	1445	1449	1452	1455	1459	1462	1466	1469	1472	1476	0	1	1	1	2	2	2	3	3
·17	1479	1483	1486	1489	1493	1496	1500	1503	1507	1510	0	1	1	1	2	2	2	3	3
·18	1514	1517	1521	1524	1528	1531	1535	1538	1542	1545	0	1	1	1	2	2	2	3	3
19	1549	1552	1556	1560	1563	1567	1570	1574	1578	1581	0	1	1	1	2	2	3	3	3
20	1585	1589	1592	1596	1600	1603	1607	1611	1614	1618	0	1	1	1	2	2	3	3	3
21	1622	1626	1629	1633	1637	1641	1644	1648	1652	1656	0	1	1	2	2	2	3	3	3
·22	1660	1663	1667	1671	1675	1679	1683	1687	1690	1694	0	1	1	2	2	2	3	3	3
·23	1698	1702	1706	1710	1714	1718	1722	1726	1730	1734	0	1	1	2	2	2	3	3	4
·24	1738	1742	1746	1750	1754	1758	1762	1766	1770	1774	0	1	1	2	2	2	3	3	4
25	1778	1782	1786	1791	1795	1799	1803	1807	1811	1816	0	1	1	2	2	2	3	3	4
·26	1820	1824	1828	1832	1837	1841	1845	1849	1854	1858	0	1	1	2	2	3	3	3	4
·27	1862	1866	1871	1875	1879	1884	1888	1892	1897	1901	0	1	1	2	2	3	3	3	4
·28	1905	1910	1914	1919	1923	1928	1932	1936	1941	1945	0	1	1	2	2	3	3	4	4
·29	1950	1954	1959	1963	1968	1972	1977	1982	1986	1991	0	1	1	2	2	3	3	4	4
30	1995	2000	2004	2009	2014	2018	2023	2028	2032	2037	0	1	1	2	2	3	3	4	4
31	2042	2046	2051	2056	2061	2065	2070	2075	2080	2084	0	1	1	2	2	3	3	4	4
·32	2089	2094	2099	2104	2109	2113	2118	2123	2128	2133	0	1	1	2	2	3	3	4	4
·33	2138	2143	2148	2153	2158	2163	2168	2173	2178	2183	0	1	1	2	2	3	3	4	4
·34	2188	2193	2198	2203	2208	2213	2218	2223	2228	2234	1	1	2	2	3	3	4	4	5
·35	2239	2244	2249	2254	2259	2265	2270	2275	2280	2286	1	1	2	2	3	3	4	4	5
·36	2291	2296	2301	2307	2312	2317	2323	2328	2333	2339	1	1	2	2	3	3	4	4	5
·37	2344	2350	2355	2360	2366	2371	2377	2382	2388	2393	1	1	2	2	3	3	4	4	5
·38	2399	2404	2410	2415	2421	2427	2432	2438	2443	2449	1	1	2	2	3	3	4	4	5
·39	2455	2460	2466	2472	2477	2483	2489	2495	2500	2506	1	1	2	2	3	3	4	5	5
·40	2512	2518	2523	2529	2535	2541	2547	2553	2559	2564	1	1	2	2	3	4	4	5	5
41	2570	2576	2582	2588	2594	2600	2606	2612	2618	2624	1	1	2	2	3	4	4	5	5
·42	2630	2636	2642	2649	2655	2661	2667	2673	2679	2685	1	1	2	2	3	4	4	5	6
43	2692	2698	2704	2710	2716	2723	2729	2735	2742	2748	1	1	2	3	3	4	4	5	6
·44	2754	2761	2767	2773	2780	2786	2793	2799	2805	2812	1	1	2	3	3	4	4	5	6
·45	2818	2825	2831	2838	2844	2851	2858	2864	2871	2877	1	1	2	3	3	4	5	5	6
46	2884	2891	2897	2904	2911	2917	2924	2931	2938	2944	1	1	2	3	3	4	5	5	6
·47	2951	2958	2965	2972	2979	2985	2992	2999	3006	3013	1	1	2	3	3	4	5	5	6
·48	3020	3027	3034	3041	3048	3055	3062	3069	3076	3083	1	1	2	3	4	4	5	6	6
·49	3090	3097	3105	3112	3119	3126	3133	3141	3148	3155	1	1	2	3	4	4	5	6	6

TABLE OF ANTILOGARITHMS.—*Continued.*

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
·50	3162	3170	3177	3184	3192	3199	3206	3214	3221	3228	1	1	2	3	4	4	5	6	7
·51	3236	3243	3251	3258	3266	3273	3281	3289	3296	3304	1	2	2	3	4	5	5	6	7
·52	3311	3319	3327	3334	3342	3350	3357	3365	3373	3381	1	2	2	3	4	5	5	6	7
·53	3388	3396	3404	3412	3420	3428	3436	3443	3451	3459	1	2	2	3	4	5	6	6	7
·54	3467	3475	3483	3491	3499	3508	3516	3524	3532	3540	1	2	2	3	4	5	6	6	7
·55	3548	3556	3565	3573	3581	3589	3597	3606	3614	3622	1	2	2	3	4	5	6	7	7
·56	3631	3639	3648	3656	3664	3673	3681	3690	3698	3707	1	2	3	3	4	5	6	7	8
·57	3715	3724	3733	3741	3750	3758	3767	3776	3784	3793	1	2	3	3	4	5	6	7	8
·58	3802	3811	3819	3828	3837	3846	3855	3864	3873	3882	1	2	3	4	4	5	6	7	8
·59	3890	3899	3908	3917	3926	3936	3945	3954	3963	3972	1	2	3	4	5	5	6	7	8
·60	3981	3990	3999	4009	4018	4027	4036	4046	4055	4064	1	2	3	4	5	6	6	7	8
·61	4074	4083	4093	4102	4111	4121	4130	4140	4150	4159	1	2	3	4	5	6	7	8	9
·62	4169	4178	4188	4198	4207	4217	4227	4236	4246	4256	1	2	3	4	5	6	7	8	9
·63	4266	4276	4285	4295	4305	4315	4325	4335	4345	4355	1	2	3	4	5	6	7	8	9
·64	4365	4375	4385	4395	4406	4416	4426	4436	4446	4457	1	2	3	4	5	6	7	8	9
·65	4467	4477	4487	4498	4508	4519	4529	4539	4550	4560	1	2	3	4	5	6	7	8	9
·66	4571	4581	4592	4603	4613	4624	4634	4645	4656	4667	1	2	3	4	5	6	7	9	10
·67	4677	4688	4699	4710	4721	4732	4742	4753	4764	4775	1	2	3	4	5	7	8	9	10
·68	4786	4797	4808	4819	4831	4842	4853	4864	4875	4887	1	2	3	4	6	7	8	9	10
·69	4898	4909	4920	4932	4943	4955	4966	4977	4989	5000	1	2	3	5	6	7	8	9	10
·70	5012	5023	5035	5047	5058	5070	5082	5093	5105	5117	1	2	4	5	6	7	8	9	11
·71	5129	5140	5152	5164	5176	5188	5200	5212	5224	5236	1	2	4	5	6	7	8	10	11
·72	5248	5260	5272	5284	5297	5309	5321	5333	5346	5358	1	2	4	5	6	7	9	10	11
·73	5370	5383	5395	5408	5420	5433	5445	5458	5470	5483	1	3	4	5	6	8	9	10	11
·74	5495	5508	5521	5534	5546	5559	5572	5585	5598	5610	1	3	4	5	6	8	9	10	12
·75	5623	5636	5649	5662	5675	5689	5702	5715	5728	5741	1	3	4	5	7	8	9	10	12
·76	5754	5768	5781	5794	5808	5821	5834	5848	5861	5875	1	3	4	5	7	8	9	11	12
·77	5888	5902	5916	5929	5943	5957	5970	5984	5998	6012	1	3	4	5	7	8	10	11	12
·78	6026	6039	6053	6067	6081	6095	6109	6124	6138	6152	1	3	4	6	7	8	10	11	13
·79	6166	6180	6194	6209	6223	6237	6252	6266	6281	6295	1	3	4	6	7	9	10	11	13
·80	6310	6324	6339	6353	6368	6383	6397	6412	6427	6442	1	3	4	6	7	9	10	12	13
·81	6457	6471	6486	6501	6516	6531	6546	6561	6577	6592	2	3	5	6	8	9	11	12	14
·82	6607	6622	6637	6653	6668	6683	6699	6714	6730	6745	2	3	5	6	8	9	11	12	14
·83	6761	6776	6792	6808	6823	6839	6855	6871	6887	6902	2	3	5	6	8	9	11	13	14
·84	6918	6934	6950	6966	6982	6998	7015	7031	7047	7063	2	3	5	6	8	10	11	13	15
·85	7079	7096	7112	7129	7145	7161	7178	7194	7211	7228	2	3	5	7	8	10	12	13	15
·86	7244	7261	7278	7295	7311	7328	7345	7362	7379	7396	2	3	5	7	8	10	12	13	15
·87	7413	7430	7447	7464	7482	7499	7516	7534	7551	7568	2	3	5	7	9	10	12	14	16
·88	7586	7603	7621	7638	7656	7674	7691	7709	7727	7745	2	4	5	7	9	11	12	14	16
·89	7762	7780	7798	7816	7834	7852	7870	7889	7907	7925	2	4	5	7	9	11	13	14	16
·90	7943	7962	7980	7998	8017	8035	8054	8072	8891	8110	2	4	6	7	9	11	13	15	17
·91	8128	8147	8166	8185	8204	8222	8241	8260	8279	8299	2	4	6	8	9	11	13	15	17
·92	8318	8337	8356	8375	8395	8414	8433	8453	8472	8492	2	4	6	8	10	12	14	15	17
·93	8511	8531	8551	8570	8590	8610	8630	8650	8670	8690	2	4	6	8	10	12	14	16	18
·94	8710	8730	8750	8770	8790	8810	8831	8851	8872	8892	2	4	6	8	10	12	14	16	18
·95	8913	8933	8954	8974	8995	9016	9036	9057	9078	9099	2	4	6	8	10	12	15	17	19
·96	9120	9141	9162	9183	9204	9226	9247	9268	9290	9311	2	4	6	8	11	13	15	17	19
·97	9333	9354	9376	9397	9419	9441	9462	9484	9506	9528	2	4	7	9	11	13	15	17	20
·98	9550	9572	9594	9616	9638	9661	9683	9705	9727	9750	2	4	7	9	11	13	16	18	20
·99	9772	9795	9817	9840	9863	9886	9908	9931	9954	9977	2	5	7	9	11	14	16	18	20

TABLE OF FUNCTIONS OF ANGLES.

Angle.		Chord	Sine.	Tangent.	Co-tangent	Cosine.			
Degrees	Radians								
0°	0	000	0	0	∞	1	1 414	1 5708	90°
1	·0175	017	·0175	·0175	57 2900	9998	1 402	1 5533	89
2	0349	035	·0349	0349	28 6363	9994	1 389	1 5359	88
3	·0524	·052	·0523	0524	19 0811	·9986	1 377	1·5184	87
4	·0698	·070	·0698	0699	14 3007	·9976	1 364	1·5010	86
5	·0873	087	·0872	·0875	11 4301	·9962	1 351	1·4835	85
6	·1047	·105	·1045	·1051	9 5144	·9945	1 338	1·4661	84
7	·1222	122	·1219	·1228	8 1443	·9925	1 325	1·4486	83
8	·1396	·140	·1392	·1405	7 1154	9903	1 312	1·4312	82
9	·1571	·157	·1564	·1584	6 3138	9877	1 299	1 4137	81
10	·1745	·174	·1736	1763	5 6713	·9848	1 286	1 3963	80
11	·1929	·192	·1908	·1944	5 1446	·9816	1 272	1 3788	79
12	·2094	·209	2079	·2126	4 7046	·9781	1 259	1 3614	78
13	·2269	·226	·2250	2309	4 3315	·9744	1 245	1·3439	77
14	2443	·244	·2419	·2493	4 0108	9703	1 231	1 3265	76
15	·2618	·261	·2588	·2679	3 7321	·9659	1·218	1 3090	75
16	2793	278	2756	2867	3 4874	9613	1 204	1 2915	74
17	·2967	·296	·2924	·3057	3 2709	·9563	1·190	1 2741	73
18	3142	313	·3090	3249	3 0777	·9511	1 176	1·2566	72
19	·3316	330	·3256	3143	2 9042	9155	1 161	1 2392	71
20	3491	347	3420	·3640	2 7475	9397	1·147	1 2217	70
21	·3665	·364	·3584	3839	2 6051	9336	1 133	1 2043	69
22	·3840	382	3746	·4040	2 4751	9272	1 118	1 1868	68
23	·4014	·399	3907	·4245	2 3559	9205	1 104	1 1694	67
24	4189	416	·4067	4452	2 2460	·9135	1 089	1 1519	66
25	·4363	·433	·4226	4663	2 1445	9063	1 075	1 1345	65
26	·4538	450	·4384	4877	2 0503	·8988	1 060	1·1170	64
27	·4712	467	·4540	·5095	1 9626	8910	1 045	1 0996	63
28	·4887	·484	4695	5317	1 8807	8829	1 030	1·0821	62
29	·5061	501	·4818	·5543	1·8040	·8746	1 015	1·0647	61
30	·5236	·518	5000	·5774	1 7321	8660	1 000	1·0472	60
31	5411	534	5150	6009	1 6643	8572	·985	1·0297	59
32	·5585	551	5299	6249	1 6003	8480	·970	1·0123	58
33	·5760	568	·5416	6494	1 5399	·8387	·954	·9948	57
34	·5934	585	5592	6745	1 4826	8290	939	·9774	56
35	·6109	·601	·5736	·7002	1 4281	·8192	·923	·9599	55
36	6283	·618	·5878	·7265	1 3764	8090	·908	·9425	54
37	·6458	635	·6018	·7536	1 3270	·7986	·892	·9250	53
38	·6632	·651	·6157	·7813	1 2799	·7880	·877	·9076	52
39	·6807	·668	·6293	·8098	1 2349	·7771	·861	·8901	51
40	·6981	·684	·6428	·8391	1 1918	·7660	·845	·8727	50
41	·7156	·700	6561	·8693	1 1504	·7547	·829	·8552	49
42	·7330	·717	·6691	·9004	1·1106	·7431	813	·8378	48
43	·7505	·733	·6820	·9325	1·0724	·7314	·797	·8203	47
44	·7679	·749	·6947	·9657	1·0355	·7193	·781	·8029	46
45°	·7854	·765	·7071	1 0000	1 0000	7071	·765	·7854	45°
			Cosine	Co tangent	Tangent.	Sine.	Chord.	Radians.	Degrees.
								Angle.	

The Institution of Civil Engineers.

EXTRACTS FROM RULES AND SYLLABUS OF EXAMINATIONS FOR

ELECTION OF ASSOCIATE MEMBERS.

PART II *—*Scientific Knowledge.*

SECTION A.

1. **Mechanics** (one Paper, *time allowed*, 3 *hours*).

2. **Strength and Elasticity of Materials** (one Paper, *time allowed*, 3 *hours*).

3. *Either* (*a*) **Theory of Structures**,

or (*b*) **Theory of Electricity and Magnetism** (one Paper, *time allowed*, 3 *hours*).

SECTION B.

Two of the following nine subjects not more than one from any group (one Paper in each subject taken, *time allowed*, 3 *hours for each Paper*):—

Group i.	*Group* ii.	*Group* iii.
Geodesy.	Hydraulics.	Geology and Mineralogy.
Theory of Heat Engines.	Theory of Machines.	Stability and Resistance of Ships
Thermo- and Electro-Chemistry.	Metallurgy.	Applications of Electricity.

* Candidates may offer themselves for examination in Sections A and B of Part II. together; or they may enter for Section A alone, and, if successful, may take Section B at a subsequent examination. In the latter case, however, such candidates will not be allowed to present themselves for examination in Section B unless or until they are actually occupied in work as pupils or assistants to practising Engineers. The Council may permit Candidates who have attempted the whole of Part II. at one examination, and have failed in Section B only, to complete their qualification by passing in that section at a subsequent examination, subject to their being then occupied as above stated.

Mathematics.—The standard of Mathematics required for the Papers in Part II. of the examination is that of the mathematical portion of the Examination for the Admission of Students, though questions may be set involving the use of higher Mathematics.

The range of the examinations in the several subjects, in each of which a choice of questions will be allowed, is indicated generally hereunder:—

SECTION A.

1. Mechanics:—

Statics.—Forces acting on a rigid body; moments of forces, composition, and resolution of forces; couples, conditions of equilibrium, with application to loaded structures. The foregoing subjects to be treated both graphically and by aid of algebra and geometry.

Hydrostatics.—Pressure at any point in a gravitating liquid; centre of pressure on immersed plane areas; specific gravity.

Kinematics of Plane Motion.—Velocity and acceleration of a point; instantaneous centre of a moving body.

Kinetics of Plane Motion.—Force, mass, momentum, moment of momentum, work, energy, their relation and their measure; equations of motion of a particle; rectilinear motion under the action of gravity; falling bodies and motion on an inclined plane; motion in a circle; centres of mass and moments of inertia; rotation of a rigid body about a fixed axis; conservation of energy.

2. Strength and Elasticity of Materials:—

Physical properties and elastic constants of cast iron, wrought iron, steel, timber, stone, and cement; relation of stress and strain, limit of elasticity, yield point, Young's modulus; coefficient of rigidity; extension and lateral contraction; resistance within the elastic limit in tension, compression, shear and torsion; thin shells; strength and deflection in simple cases of bending; beams of uniform resistance; suddenly applied loads.

Ultimate strength with different modes of loading; plasticity, working stress; phenomena in an ordinary tensile test; stress-strain diagram; elongation and contraction of area; effects of hardening, tempering and annealing; fatigue of metals; measurement of hardness.

Forms and arrangements of testing machines for tension, compression, torsion, and bending tests; instruments for measuring extension, compression, and twist; forms of test pieces and arrangements for holding them; influence of form on strength and elongation; methods of ordinary commercial testing; percentage of elongation and contraction of area; test conditions in specifications for cast iron, mild steel, and cement.

3. (a) Theory of Structures:—

Graphic and analytic methods for the calculation of bending moments and of shearing forces, and of the stresses in individual members of framework structures loaded at the joints; plate and box girders; incomplete and redundant frames; stresses suddenly applied, and effects of impact;

buckling of struts; effect of different end fastenings on their resistance; combined strains; calculations connected with statically indeterminate problems, as beams supported at three points, &c.; travelling loads; riveted and pin-joint girders; rigid and hinged arches; strains due to weight of structures; theory of earth-pressure and of foundations; stability of masonry and brickwork structures.

3. (*b*) Theory of Electricity and Magnetism:—

Electrical and magnetic laws, units, standards, and measurements; electrical and magnetic measuring instruments; the theory of the generation, storage, transformation, and distribution of electrical energy; continuous and alternating currents; arc and incandescent lamps; secondary cells.

SECTION B.

Group i. Theory of Heat Engines:—

Thermodynamic laws; internal and external work; graphical representation of changes in the condition of a fluid; theory of heat engines working with a perfect gas; air- and gas-engine cycles; reversibility, conditions necessary for maximum possible efficiency in any cycle; properties of steam; the Carnot and Clausius cycles; entropy and entropy-temperature diagrams, and their application in the study of heat engines; actual heat engine cycles and their thermodynamic losses; effects of clearance and throttling; initial condensation; testing of heat engines, and the apparatus employed; performances of typical engines of different classes; efficiency.

Group ii. Hydraulics:—

The laws of the flow of water by orifices, notches, and weirs; laws of fluid friction; steady flow in pipes or channels of uniform section; resistance of valves and bends; general phenomena of flow in rivers; methods of determining the discharge of streams; tidal action; generation and effect of waves; impulse and reaction of jets of water; transmission of energy by fluids; principles of machines acting by weight, pressure, and kinetic energy of water; theory and structure of turbines and pumps.

Theory of Machines:—

Kinematics of machines; inversion of kinematic chains; virtual centres; belt, rope, chain, toothed and screw gearing; velocity, acceleration and effort diagrams; inertia of reciprocating parts; elementary cases of balancing; governors and flywheels; friction and efficiency; strength and proportions of machine parts in simple cases.

Group iii. Applications of Electricity:—

Theory and design of continuous and alternating-current generators and motors, synchronous and induction motors and static transformers; design

of generating- and sub-stations and the principal plant required in them; the principal systems of distributing electrical energy, including the arrangement of mains and feeders; estimation of losses and of efficiency; principal systems of electric traction; construction and efficiency of the principal types of electric lamps.

☞ Candidates should see, that all their "Forms" are *duly completed* and *passed* by the Council of the Institution of Civil Engineers, Great George Street, Westminster, S W., *before* 1st January for the February Examination, and *before* the 1st September for the October Examination

Examinations Abroad.—The papers of the *October* Examination *only* will be placed before accepted Candidates in India and the Colonies. To enable the Secretary to make arrangements for the Application Forms and Fees, &c of these Candidates, their Forms, &c., *must* be in the Secretary's hands, *before* the 1st June preceding the October Examinations.

THE INSTITUTION OF CIVIL ENGINEERS
EXAMINATION, OCTOBER, 1913.

ELECTION OF ASSOCIATE MEMBERS.

THEORY OF HEAT ENGINES

Not more than four questions to be attempted by any Candidate.

1. State briefly the advantages of using multi-stage air compressors. Air is compressed adiabatically from a pressure of 15 lbs. per square inch absolute to 90 lbs. per square inch absolute. Find the final temperature of the air if the initial temperature is 60° F. (Assume $\gamma = 1.4$.)

2. The following results were obtained from the test of a gas engine working on the Otto cycle:—

Duration of test in minutes, ...	90
Indicated horse power, ...	150
Total gas used in cubic feet . . .	3,825
Calorific value of gas per cubic foot B.Th.U. . . .	550
Clearance volume 15 per cent. of piston displacement	

Find the thermal efficiency and the efficiency-ratio of the engine.

3. Sketch and describe some system of forced draught for steam boilers, and state briefly the advantages of forced draught, and its effects upon the thermal efficiency of boilers.

4. Steam enters the nozzle of a De Laval turbine and expands to the condenser pressure. The theoretical heat drop is 230 B.Th.U. per pound, but 10 per cent. of the energy is lost in friction. Draw the velocity-diagram of the steam passing through the turbine if the relative velocity at exit is 85 per cent. of the inlet velocity. Find the efficiency of the nozzle and vanes, assuming the steam enters the vanes without shock and that the inlet and outlet angles of the vanes are equal. The peripheral velocity of the wheel is 1,200 feet per second. The nozzles are inclined at an angle of 20° to the plane of the wheel.

5. The following results were obtained from a test of a steam engine controlled by a throttling governor:—

Indicated Horse Power	Lbs. of Steam per Hour
103	1,615
304	4,630

Assuming Willans's law, find an equation connecting the horse-power and the steam used; also find the steam consumption per hour at 220 I.H.P.

6. Sketch the temperature-entropy diagram for ammonia, and show on it the refrigerating cycle for wet compression How is the coefficient of performance determined?

7. Find the work done per pound of steam by a steam engine working on the Rankine cycle between 362° F and 152° F. How many pounds of steam are required per horse-power, given the following Table?

Temperature, ° F	Entropy of 1 Lb of Water	Total Entropy of 1 Lb. of Steam.
362	0·519	1·566
152	0·218	1·863

8. Find the dryness of the steam after cut-off as three-quarters of the stroke from the following particulars of an engine-trial, assuming no leakage:

Condensed steam per hour in pounds, . .	4,608
Revolutions per minute, ...	120
Volume of cylinder, cubic feet, ...	3·6
Clearance per cent , ...	5
Pressure of steam in pounds per square inch as $\frac{3}{4}$ stroke,	41·8
Volume in cubic feet of 1 lb of steam at 41·8 lbs pressure,	10·05
Pressure of steam in pounds per square inch at 0 84 of the return stroke and commencement of compression,	17·2
Volume in cubic feet of 1 lb of steam at 17·2 lbs per square inch, ...	23·14

9. Describe with a sketch (i) some method of measuring high temperature in heat engines; or (ii) some method of measuring the fluctuation of speed of rotation of an engine

10. Describe the Ideal Carnot cycle by means of a $p\,v$ diagram, and by means of a temperature-entropy diagram A perfect engine working on the Carnot cycle receives 5,000 B Th U per minute at 2,000° F.; the heat is rejected at 500° F. Find the horse-power and the efficiency of the engine.

11. Explain with a sketch the working of a suction gas producer. What is the object of admitting vapour with the air? Describe any method of regulating the vapour supply

12. Steam passes through a throttling calorimeter where it is reduced in pressure from 120 lbs per square inch (temperature 341° F.) to 15 lbs. per square inch. The temperature after expansion is 230° F. The temperature of steam at 15 lbs per square inch is normally 213° F Find the original dryness of the steam The latent heat of 1 lb of dry steam is approximately 1,114 – 0·7 t thermal units, where t is the temperature of the steam in degrees Fahrenheit. The specific heat of superheated steam may be taken as 0·5.

February, 1914.

THEORY OF HEAT ENGINES

Not more than EIGHT *questions to be attempted by any Candidate.*

1. Find the diameter of the high- and low-pressure cylinders of a compound engine to develop 120 indicated horse-power with a piston speed of 600 feet per minute. Ratio of cylinder volumes 1 : 3·25 Admission pressure, 115 lbs per square inch absolute, condenser pressure, 3 lbs. per square inch absolute; cut-off in the high pressure cylinder, 0·5. Assume a diagram factor of 0 7

2 Sketch and describe some form of carburettor for a petrol engine. State some of the difficulties to be overcome in the design of carburettors for motor work

3. Air is drawn into a compressor at atmospheric pressure and compressed to a pressure of five atmospheres Find the horse-power required to compress and deliver 1,000 cubic feet of free air, assuming (*a*) isothermal compression, (*b*) adiabatic compression ($\gamma = 1 4$)

4. The following results were obtained from the test of a steam engine:—

Mean pressure on the piston in pounds per square inch,	40
Revolutions per minute (double-acting),	180
Net load on brake in pounds,	400
Radius of brake load in feet,	5
Steam condensed per hour in pounds,	1,256
Diameter of cylinder in inches,	12
Length of stroke in feet,	1·5

Find the indicated horse-power, brake horse-power, mechanical efficiency, and steam per horse-power-hour

5. Use the following Table to draw a temperature-entropy diagram, and from it find the work done per pound of dry steam by a perfect engine working on the Rankine cycle between 150 lbs per square inch and 16 lbs. per square inch absolute pressures Also determine the dryness of the steam after expansion.

Pressure. Lbs. per Square Inch	Temperature ° F.	Entropy of 1 Lb of Water.	Total Entropy of 1 Lb of Steam.
150	359	0·514	1·569
16	216	0·319	1·749

6. Make an outline sketch of either a link-motion, or of some form of radial valve-gear. Explain how you would determine the angle of advance and eccentricity of the equivalent eccentric for a given position of the gear.

7 State the chief items of cost in generating a unit of energy: (*a*) by steam-plant, (*b*) by gas-plant in a power station. What is the effect of the load-factor on these items, and how is the question of the type and dimensions of the plant affected by the load-factor?

8. The following results were obtained from tests of two separate boilers:—

	No. I.	No. II.
Water evaporated per pound of coal, lbs .	8·7	9·2
Temperature of feed-water, ° F,	60	200
Temperature of saturated steam at boiler-pressure, ° F.,	340	380
Degrees of superheat, F.,	..	100
Dryness of steam,	0·98	..

Assuming the quality of the coal to be the same in both cases, determine which is the more efficient boiler Latent heat of 1 lb. of dry steam = 1,114 − 0·7 *t*, where *t* is the temperature of the steam in ° F. The specific heat of superheated steam may be taken as 0·5.

9. Describe briefly the principle of action of De Laval, Rateau, Curtis, and Parsons steam turbines. Show by a diagram how the pressure and the velocity of the steam vary in passing through any one of these turbines.

10. Explain fully the reason of the efforts made to improve the quality of the vacuum in steam-turbine plants, and describe some form of improved condensing and air-pump plant used for this purpose

11. The volumetric analysis of the flue-gas of a boiler gave the following results:—CO_2, 10 per cent ; oxygen, 9·7 per cent.; nitrogen, 80·3 per cent. Find the weight of air supplied per pound of coal if the coal contained 82 per cent. of carbon, 5·3 per cent. of hydrogen, 4 per cent. of ash, and 8·7 per cent. of incombustible gaseous products.

12. The pressure and volume at several points in the compression of a gaseous mixture in a gas-engine cylinder is shown in the following Table:—

Pressure,	20	45	80	130
Volume,	4·73	2·52	1·62	1·10

Assume the curve can be represented by an equation of the form $P V_n = c$, and determine the value of *n*.

October, 1914.

THEORY OF HEAT ENGINES.

Not more than EIGHT *questions to be attempted by any Candidate*

1. State the second law of Thermodynamics and show how the growth of entropy in irreversible cycles is in conformity with this law.

2. Define the following terms: specific heat of a gas at constant volume; adiabatic expansion; latent heat of evaporation; total heat of a saturated vapour; heat of a liquid. State the relation existing between the last three terms for the same liquid.

3. What is the difference between the theoretical Otto and Diesel cycles? Which is theoretically the most efficient, assuming the same ratio of compression in both cases? Which is practically the most effieient, and why?

4. What is meant by a reversible cycle and by an irreversible cycle? Which portions of the cycle of an actual simple steam engine are irreversible?

5. Draw a Mollier heat-chart with the date given below (only the 200° F. superheat line, the saturation line, and the lines representing the given pressures need be drawn). Why are the pressure lines straight in the saturated field and curved in the superheated field? Assuming the initial condition of the steam to be 200 lbs per square inch absolute pressure and 200° F. superheat, draw a line representing the Rankine cycle and a line giving 70 per cent efficiency ratio; in both cases terminating at 2 lbs. absolute pressure.

	Pressure.	Total Heat.	Entropy.
	Lbs per Sq In abs	B Th U per Lb	
Superheat 200° F.,	200	1,308	1·663
	100	1,289	1·719
	25	1,256	1·833
	5	1,222	1·972
	1	1,206	2·053
Saturated,	200	1,198	1·546
	100	1,186	1·602
	25	1,160	1·714
	5	1,130	1·843
	1	1,115	1·918
0·8 dryness fraction,	200	1,030	1·346
	100	1,009	1·377
	25	970	1·444
	5	930	1·520
	1	911	1·570

6. What is meant by the term "missing quantity" in the case of a steam engine? Give your views as to the relative effect of valve leakage, cylinder walls, and moisture film on the missing quantity.

7. Describe and illustrate with hand-sketches a method of weighing the feed water in a boiler trial What kind of corrections have to be made to the feed so measured to obtain the actual rate of feed of the boiler?

8. Describe one of the following kinds of brakes:—(*a*) Prony brake; (*b*) Froude water-brake. Obtain from first principles the formula for calculating the B.H.P.

9. The economy of a steam engine is generally stated as the number of pounds of steam required per I.H.P.-hour. Show that this statement

is incorrect and that the error is greater with superheated than with saturated steam.

10. In the Diesel type of oil engine a portion of the horse-power developed in the cylinders is required for working the air-compressor. Should this fact be taken into account when calculating the oil consumption per Indicated H.P., and if so in what way and to what extent?

11. One pound of air is compressed adiabatically to one-fourth of its original volume. Calculate the temperature reached, taking the initial temperature at 60° F. and $y = 1{\cdot}4$. How many B.Th U. must be removed from the compressed air in order that it may be cooled to the original temperature, without changing its compressed volume? $C_v = 0{\cdot}17$.

12. The dryness of fraction of steam is ascertained to be 0·96 and the pressure is 170 lbs per square inch absolute. Find the total heat of this steam given that the total heat of saturated steam at this pressure is 1,195·4 B.Th.U. per pound, and that the water heat is 340·7 B.Th.U. per pound. If this steam were expanded by throttling, show how to calculate the temperature when the steam is just saturated.

February, 1915.

THEORY OF HEAT ENGINES.

Not more than EIGHT *questions to be attempted by any Candidate*

1. Explain the statement that if the condition of a substance is changed along a reversible path the difference of entropy between the final and the initial stages is the summation of $\frac{dQ}{T}$. Show that the difference of entropy is greater than this if the path is irreversible.

2. Obtain the usual expression for the adiabatic expansion of a perfect gas.

3. Assuming constant specific heat, *sketch* the theoretical Otto cycle both on the $p\,v$ and on the $\theta\,\varphi$ diagram, and explain the meaning of each line. Superimpose in each case diagrams that might be obtained from an actual engine, stating the causes of the differences between the actual and the theoretical diagrams.

4 *Sketch* a θ-φ chart for steam and show by its means that the loss due to reducing the admission pressure by throttling is considerably less than that caused by an equal increase in back-pressure.

5. Describe shortly how you would carry out the test of a gas engine of, say, 20 B.H.P using town's gas with the object of ascertaining the gas consumption per B.H.P.-hour.

6. Given the analysis of the fuel, of the flue gases, the temperature of the flue gases leaving the boiler and the temperature of the air in the boiler room, explain the method of calculating the heat carried away by the flue gases per pound of fuel.

7. The conditions under which a steam engine is working give an available heat fall of 318 B Th.U per lb. of steam The result of a test gave a steam consumption of 10 lbs. per I.H P.-hour. Calculate the efficiency ratio Does the result throw any doubt on the accuracy of the test of steam consumption?

8. The thermal efficiency of a gas engine working with suction gas is 30 per cent The efficiency of the producer is 80 per cent, and anthracite of 14,000 B.Th.U. per lb is used, costing 28s per ton The overall thermal efficiency of a steam plant is 15 per cent using coal of 12,500 B Th U. per lb, costing 12s. per ton The brake efficiencies are 86 and 92 per cent. for the gas and the steam engine respectively What is the cost of fuel in each case working at 100 B H P. for 3,000 hours, omitting standby losses?

9. Explain the terms specific heat at constant volume, and specific heat at constant pressure What views are now held as to the variation of these specific heats with temperature and with pressure in respect of the products of combustion of a gas engine?

10. The ratio of the high-pressure to the low-pressure cylinder of a double-acting compound steam engine is 1 to 3 The average mean pressure of the high-pressure diagrams is 51 lbs per square inch and that of the low-pressure diagrams 15 lbs per square inch The area of the low-pressure piston is 700 square inches, the stroke is 2 feet, and the speed 180 revolutions per minute. Calculate the I H.P. If the B H P measured is 450, what horse-power is required to overcome engine friction, and what is the brake efficiency?

11. The explosive mixture in a gas-engine cylinder can produce 600 B Th.U. per lb. At the beginning of compression the temperature is 250° F. Compression takes place adiabatically to one-seventh the original volume Explosion then occurs at constant volume What theoretical temperature would be reached, assuming constant specific heats. $C_v = 0{\cdot}19$ and $C_p = 0{\cdot}26$?

12. Describe some form of throttling calorimeter for determining the dryness fraction of steam, explaining the principles of its action and stating the measurements to be taken

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October, 1915.

THEORY OF HEAT ENGINES.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. What is meant, in thermodynamics, by the terms "reversible" and "irreversible" operations? Show that, within the same temperature limits, no engine can be more efficient than a reversible engine

2. Prove the equation $V = w + J\dfrac{L}{\tau}\dfrac{d\tau}{dp}$, by which the volume per pound of dry saturated steam may be deduced from a knowledge of the latent

heat—temperature and the pressure—temperature curves and the density of water. Calculate the volume per pound of dry saturated steam at 200 lbs. per square inch absolute from the following data :—

Pressure (lbs. per square inch absolute), .	195	200	205
Temperature, ° F., ...	379·4	381·6	383·7

The latent heat as pressure 200 is 850·3 B Th U.

3. In respect to engine indicators of the ordinary type, enumerate the chief sources of error arising in connection with the attachment and adjustment of the instrument State how the errors can be minimised.

4. Air, at a temperature of 80° F., is compressed from 15 to 90 lbs. per square inch absolute, and the curve of compression is $p\,v^{1\cdot 2}$ = constant. Find (1) the work done, and (2) the heat lost to the cylinder walls, per pound of air. The specific heats at constant pressure and constant volume are 0·2375 and 0·169 respectively.

5. In a surface condenser the tubes were $\frac{1}{20}$ inch thick, and the thermal conductivity of the material was such that 25 B Th U. were transmitted across a plate 1 inch thick per square foot per hour per degree difference of temperature of the two sides of the plate Calculate the heat flow across the tubes if the outside and inside were at the temperatures of the steam (132° F.) and water (80° F.) respectively. Explain why the actual heat flow is smaller than that calculated

6. In an ideal engine cycle with constant admission and back pressure and curves of expansion and compression $p\,v^n$ = constant; show that the amount of clearance has no effect upon the efficiency, provided the expansion is carried to the lower and the compression to the higher pressure. What are the effects of clearance and compression in ordinary engines?

7. In an engine working on the Rankine-Clausius cycle, the steam being dry saturated at entry, the limits of temperature are 344° F. (latent heat 880 B.Th.U.) and 126° F. (latent heat 1,020 B.Th U.); the thermal efficiency is 25 per cent. If the steam is superheated 200° F. at inlet, calculate the additional work done and the thermal efficiency. Explain why the addition of heat as superheat gives proportionately more work (1) in theory, (2) in practice

8. A diverging nozzle is supplied with superheated steam at a given pressure, and expands the steam to a lower pressure at which the quality is given. Show how to calculate the diameter at the throat and at exit for a given discharge and how to determine the velocity of the steam at exit.

9. A triple-expansion marine engine is required to give 2,000 I.H.P., the piston speed being 720 feet per minute, and the cut-off being at 0·7 of the stroke in the H P. cylinder. The steam-chest and back pressures are 205 and 2 lbs. per square inch respectively. Taking the diagram factor as 0·65 and the ratio of the L P. to H.P. cylinder areas as 7·8, find suitable cylinder diameters.

10. Draw p-v and θ-φ diagrams for a refrigerating machine, using air, which has admission and exhaust at constant pressures; and adiabatic compression and expansion. If the temperatures at beginning and end of compression are 20° F and 390° F, and the temperature at the beginning of expansion is 100° F., find the coefficient of performance.

11. Draw up a heat balance for the following test of a jacketed engine, expressing the results in percentage of heat supplied :—I H.P., 200; admission pressure, 150 lbs. per square inch absolute; steam 3 per cent.

wet; cylinder feed, 3,550 lbs. per hour; jacket feed, 200 lbs. per hour; circulating water, 66,000 lbs. per hour, the rise of temperature being 49° F.; feed temperature, 120° F. At 150 lbs pressure the sensible and latent heats are 331 and 869 B.Th U per pound.

12. In a gas engine working on the Otto cycle the clearance volume is one-third of the volume swept by the piston per stroke Find the thermal efficiency of the corresponding ideal air cycle Find also the mean pressure in the ideal cycle if the pressure at the beginning of compression is 15 lbs. per square inch and if the maximum pressure is 3 times the pressure at the end of compression. Take the ratio of specific heats as 1·4.

February, 1916.

THEORY OF HEAT ENGINES.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. Show how an absolute scale of temperature can be defined, and state how air and mercury thermometers differ from such a scale. Using the absolute scale, find an expression for the efficiency of a perfect heat engine.

2. Steam 5 per cent wet at pressure 105 lbs per square inch absolute is throttled to 85 lbs per square inch absolute and then expanded adiabatically to 40 lbs. per square inch absolute. Find the wetness after throttling and after expansion. At pressures 40, 85, and 105 the temperatures are 267°, 316°, and 331° F., and the latent heats are 935, 901, and 890 B.Th.U.

3. Describe with sketches *either*—(1) an engine indicator of the optical type; *or* (2) a shaft transmission dynamometer for a steam turbine

4. Prove the relation that exists between the specific heats, at constant volume and constant pressure, of a gas. Calculate the difference between the two specific heats of air, having given that a pound of air has a volume of 13·1 cubic feet at a temperature of 60° F. and pressure 14·7 lbs. per square inch absolute.

5. Discuss the results of experiments upon the rate of transmission of heat to metal surfaces from gases and water flowing at high speeds. Indicate the bearing of such experiments on boiler and condenser design.

6. Taking the Rankine-Clausius cycle as the standard of comparison for the efficiency of steam engines, enumerate the sources of loss in the actual as compared with the ideal engine. The steam consumption of an engine is 13·5 lbs. per I H.P -hour, and the work done in the corresponding Rankine-Clausius cycle is 295 B.Th U per pound. Find the efficiency ratio.

7. In a simple impulse turbine of the de Laval type, the mean blade speed is 1,200 feet per second, the angle of the jet is 20°, and the blade angles are each 36°. If there is no loss by shock at entrance, find the velocity of the steam. For that velocity find the work done on the blades per pound per second and the ratio of the work done to the kinetic energy of the steam. Take the relative velocity at exit as 90 per cent. of the relative velocity at inlet.

8. Discuss the use of superheated steam in engines from the thermo-dynamic and practical aspects Note some of the chief points to be considered in the design of engines for use with highly superheated steam.

9. Show how to combine the full load indicator cards for a triple-expansion engine, exhibiting them in relation to one saturation curve. Indicate the effect on the diagrams of throttling the steam supply.

10. Sketch a total heat-entropy chart for steam, showing curves of constant pressure, constant dryness, and constant superheat. Explain som of the uses of this chart in engine and turbine problems.

11. Exhibit on a θ-φ chart the cycle of a vapour compression refrigerat using ammonia, the compression being wet and the liquid being admitte through the expansion valve before cooling to the lower temperatur Show how to find the coefficient of performance. A plant produces 1 to of ice per hour, each pound representing 170 B Th U. If the coefficient performance is 60 per cent. of that of the corresponding ideal cycle, in whic 550 B Th U. are absorbed for an expenditure of 74 B Th U of work, fin the horse-power necessary to drive the compressor.

12. Calculate the thermal efficiency of an ideal air engine working on th Diesel cycle. The compression and expansion curves are adiabatic ($y = 1{\cdot}4$ and the ratios of compression and expansion are 14 and 7 respectively. The temperature at the beginning of compression is 140° F.

February, 1917.

THEORY OF HEAT ENGINES.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. One pound of air at 60° F is drawn into an air compressor at 15 lbs. per square inch absolute pressure, compressed adiabatically to 90 lbs. per square inch and delivered to a receiver Assuming the compressed air to be used in a motor without expansion at 60° F, find the ratio of the work done in the motor to the work expended in the compressor. Neglect losses

2. Draw the ideal indicator diagram from the following data, and find the mean effective pressure Admission pressure 80 lbs per square inch by gauge; cut-off at 0 4 of stroke; release at 0 95 of stroke; compression at 0·9 of stroke; back pressure 17 lbs per square inch absolute; clearance 6 per cent.; expansion and compression curves pv = constant.

3. A gas engine uses 19 cubic feet of gas per hour per indicated horse-power (calorific value 550 B Th U per cubic foot). Find the thermal efficiency of the engine and the efficiency ratio, if the compression ratio is 4·5 to 1. Assume the ratio of the specific heats is 1·4.

4 In a trial of a steam boiler the feed water entered the boiler at 100° F. and was evaporated into steam at 382° F The steam was afterwards raised to 582° F by the flue gases without change of pressure Find the equivalent evaporation from and at 212° F if 9 pounds of water were evaporated per pound of coal consumed. Why is the "equivalent evapora-

tion from and at 212° F" usually stated in boiler trials? Latent heat of steam = 1,114 − 0·7 t, where t is the temperature of evaporation. Specific heat of superheated steam = 0·5.

5 One pound of water at 32° F. is raised to 328° F. and then converted into steam at 328° F. Draw an entropy scale for the above process, and complete the temperature-entropy diagram. (The latent heat of one pound of steam at 328° F. = 890 B.Th.U.) State numerically the heat units represented by 1 square inch of the diagram.

6. Show approximately the positions of the crank of a gas engine when the valves open and close and when ignition takes place. How can the valve setting of a gas engine be practically determined?

7. Explain what is meant by wet and dry compression in an ammonia compressor. Illustrate your answer by reference to the temperature-entropy diagram. In a test of an ammonia compressor 80 lbs. of cooling water were required per minute, having an inlet temperature of 55° F. and an outlet temperature of 67° F. Diameter of compressor cylinder 6 inches; stroke 10 inches; 150 strokes per minute; mean pressure 55 pounds per square inch. Find the coefficient of performance, neglecting all heat losses.

8. The cut-off of the steam in a steam-engine cylinder is required to be at 0·5 of the stroke; angle of lead 6°; greatest opening to steam 2 inches. Find the travel of the valve, lap of the valve, and angle of advance of the eccentric. Neglect the obliquity of the connecting-rod.

9 Show by sketches how to take a sample of the boiler flue gases and how to determine the percentage of carbon dioxide and oxygen present. What precautions are necessary in order to obtain an accurate result? State approximately the percentage of each when the boiler is working economically with coal as fuel.

10. What is meant by the higher and lower calorific value of a fuel? Explain how the calorific value of a fuel oil may be determined. In a calorimeter experiment it was found that 0·017 lb. of oil was consumed and 6 lbs. of water were raised 54° F. Calculate the higher calorific value of the oil.

11. Explain with a sketch the construction and working of some type of gàs producer for gas engines. Compare the suction gas producer with the pressure gas producer.

12. In recent years steam turbines have been combined with reduction gearing in certain cases for the propulsion of ships. Explain briefly the reasons for such speed reduction, and sketch and describe any type which has been used.

February, 1918.

THEORY OF HEAT-ENGINES.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. State the second law of thermodynamics and apply it to define the thermal efficiency of a steam engine.

2. Make hand sketches of the indicator diagram of a simple condensing steam engine cutting off at about 0·3, also of a gas engine, choosing your own data as regards pressures. Compare the two diagrams in respect of the admission, expansion, and compression lines, stating approximately the pressures and temperatures at the beginning and end of each.

3. Explain how the ratio of the cylinders of a compound reciprocating engine can be determined, so that the horse-power in each cylinder shall be equal, using either the p.v. or the $\theta\varphi$ method.

4. What mechanical and thermodynamical advantages are gained by combining turbines of the impulse and reaction types (known as the disc and drum type)? Describe shortly the construction of such a turbine, illustrating your answer with a hand-sketch.

5. Make a hand-sketch of a longitudinal section of the cylinder of a "uniflow" steam engine. Describe the principle applied and show why this type is economical.

6. Define the following terms in relation to steam:—Latent heat, total heat, water heat: what relation exists between the three. Calculate the B.Th.U. required to produce 1 lb. of steam having a dryness fraction of 0·8 from 1 lb. of water at 100° F., given that the temperature of the steam is 350° F., and the latent heat 870 B.Th.U. per lb. at the latter temperature. Make a hand sketch of a $\theta\varphi$ diagram explanatory of the above.

7. Show that the power that can be developed by a gas engine depends to a very small extent on the calorific value of the gas.

8. It is desired to determine the gas consumption per B.H.P., the mechanical efficiency and the thermal efficiency of a small gas engine working with town gas. Give a list of the testing appliances required and describe how you would conduct the test.

9. What are the heat losses in a gas engine? Show how they can be exhibited on a $\theta\varphi$ diagram.

10. A steam turbine and a reciprocating steam engine when working with 27 inches of vacuum have the same steam consumption. Show that if the vacuum is increased, to say 29 inches, the steam consumption of the turbine will be greatly improved, but that of the reciprocating engine hardly at all.

11. Explain what is meant by clearance in a steam engine. What effect has an increase of clearance on the mean pressure, and on the steam consumption.?

12. Define the term "efficiency ratio," and state what data are required to compute it:—

(*a*) In the case of an oil engine.
(*b*) ,, ,, gas ,,

A test gave 13·4 lbs. of steam per kilowatt-hour in the case of a steam turbine when the steam conditions were such that the heat drop was 380 B.Th.U. per lb. Assuming that mechanical and electrical losses in the steam turbine and the generator are together 10 per cent., calculate the "efficiency ratio." One kilowatt-hour is the equivalent of 3,410 B.Th.U.

October, 1919.

THEORY OF HEAT ENGINES.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. Sketch pressure-volume diagrams of the following, neglecting clearance and compression :—

(*a*) An engine working on the Rankine cycle.
(*b*) „ „ non-expansively.
(*c*) „ „ with incomplete expansion—that is, the back pressure is less than the pressure at the end of expansion.

Make sketches showing the temperature-entropy diagrams for the above and cross hatch the area showing the loss due to incomplete expansion.

2. Steam leaves the nozzles of a Curtis turbine at 2,930 feet per second. Assuming the inlet and outlet angles of the blades are the same, draw the velocity diagram if the steam enters the blades without shock. Allow a loss of velocity of 10 per cent in both the moving and stationary blades. State the blade angles and the final velocity of the steam. The inlet nozzles are inclined at an angle of 20° to the plane of the wheel, and the velocity of the wheel is 450 feet per second. The turbine consists of two rows of moving blades and one row of stationary blades.

3. State briefly the thermal and mechanical advantages of two-stage air compressors. What is the maximum pressure to which you consider air should be compressed in a single stage? Give reasons for your answer. Air is compressed adiabatically from 14 lbs. per square inch absolute to 100 lbs. per square inch absolute. Find the temperature at the end of compression if the initial temperature is 60° F.

4. Describe the cycle of operations of a refrigerating machine of the compression type using ammonia as the working substance. How does the cycle differ from that of the reversed Carnot cycle? Illustrate your answer by the temperature-entropy diagram.

5. State the difficulties to be overcome in designing a carburettor for a petrol engine and make a sketch illustrating the construction of one type. Explain any special advantage of the type you select.

6. Assuming that the specific heat of a gas is constant and that the adiabatic expansion of the gas may be represented by the equation PV^n = constant; show that the index n is the ratio of the specific heat at constant pressure to the specific heat at constant volume.

7. State what you consider to be in Lancashire boiler flues, a satisfactory percentage of CO_2 and a satisfactory temperature for the chimney gases, with and without economisers. What are the most common causes of inefficiency in boiler plants and how can the efficiency be improved?

8. The lap and lead of a valve is 1 inch and cut-off takes place at 0·75 of the stroke. Find the travel of the valve and the angle of advance of the eccentric if the lead is $\frac{1}{16}$ inch. Neglect the obliquity of the connecting-rod.

9. State three of the methods which have been adopted for governing gas engines. Discuss briefly the circumstances under which each system is likely to give good results.

10. It is usual to make a special effort to obtain the highest possible vacuum in a steam turbine plant. Explain why it is more important to secure a high vacuum in a turbine plant than in the ordinary reciprocating engine plant, illustrating your answer by the temperature-entropy diagram.

11. Steam enters a cylinder at a pressure p_1 and expands according to the law P V = constant; the back pressure is p_2. Prove that the maximum work per cubic foot of steam is obtained when the number of expansions is $\frac{p_1}{p_2}$. Neglect initial condensation, compression, and clearance.

12. Make a line sketch of the Stephenson Link Motion (a) with open rods, (b) with crossed rods. State the effect on the lead and on the valve travel as the block moves from the end of the link to the centre of the link in both cases.

April, 1920.

THEORY OF HEAT ENGINES.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. A perfect steam engine works on the Rankine cycle between 300° F. and 120° F. Find the dryness of the steam after expansion, the work done per pound of steam and the pounds of steam required per hour per horse-power, given the following Table:—

Temperature ° F.	Entropy of 1 Lb. of Water.	Total Entropy of 1 Lb. of Steam.
300	0·438	1·626
120	0·165	1·941

2. In a steam turbine the velocity of the steam leaving the nozzles is 2,900 feet per second, and the nozzles are inclined at an angle of 20° to the plane of the wheel. The mean peripheral velocity of the wheel is 1,150 feet per second. Find the efficiency of the wheel, assuming the exit relative velocity from the wheel is 90 per cent. of the inlet relative velocity. Assume that the inlet and outlet angles of the blades are equal and that the steam enters the blades without shock.

3. Discuss briefly the advantages and disadvantages of any two of the working fluids used in refrigerating machines. Sketch in outline the construction of one type of refrigerating plant.

4. Air expands according to the law PV^n = constant. Find an expression for the heat added or subtracted during the expansion.

5. Make an outline sketch of some type of steam boiler and state the conditions determining the transmission of heat from a gas passing through a boiler tube to the water outside the tube.

6. In testing boilers it is usual to state the evaporation of the water as "equivalent evaporation from and at 212° F." State the reason of this. A steam boiler evaporates 8½ lbs. of water from feed water at 50° F. to steam at 340° F. per pound of coal burned. Find the equivalent evaporation from and at 212° F. if the steam is 95 per cent. dry. (Latent heat of one pound of dry steam at 340° F. is 883 B.Th.U.)

7. Find the theoretical efficiency of a gas engine using the four-stroke cycle, assuming the specific heat is constant. Find the efficiency when the volume before compression is five times the volume at the end of compression. What is the approximate efficiency for the actual case of variable specific heat?

8. Explain how to obtain the brake horse-power of an oil engine. An oil engine uses 38 lbs. of oil per hour, having a calorific value of 18,000 B.Th.U. per pound. The I.H.P. is 90·8 and the B.H.P. is 62·3. Find the mechanical efficiency and the thermal efficiency of the engine.

9. Explain with a sketch the construction of some type of suction gas producer for use with a gas engine. Show how the vapour is supplied and the gas cleaned.

10. Twenty cubic feet of air are compressed adiabatically from 15 lbs. per square inch to 90 lbs. per square inch absolute. Draw the diagram for compression and delivery of the air, and find the work done by the engine. Neglect clearance.

11. Make an outline sketch of some type of radial valve gear and from the dimensions of the gear show how to find the equivalent eccentric and its angle of advance for any position of the gear.

12. The travel of a slide valve is 4 inches and the cut-off takes place at 0·8 of the stroke. Find the steam lap and angle of advance if the lead is ¼ inch. Draw an approximate indicator diagram if the exhaust lap is 0·25 inch and the clearance 7½ per cent. Assume the initial pressure is 70 lbs. per square inch and the back pressure 16 lbs. per square inch absolute.

October, 1921.

THEORY OF HEAT ENGINES.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. Give a general definition of a heat engine. Enumerate the component parts of the elementary heat engine, and state what is meant by the "cycle." Sketch the theoretical pressure-volume and temperature-entropy diagrams for an engine using a permanent gas (*a*) when heat is received and rejected at constant volume; (*b*) received at constant pressure and rejected at constant volume. Indicate, in each case, in what respect the practicable differs from the ideal cycle.

2. A gas producer is supplied with coal having a calorific value of 13,000 B.Th.U./lbs. One ton of coal yields 160,000 feet^3 of gas, at standard

temperature and pressure (S.T.P.), which has the following composition by volume:—CO_2 17·1, CO 11·0, C_2H_4 0·4, CH_4 1·8, H_2 27·2, N_2 42·5 per cent. Calculate the air required for complete combustion of 1 cubic foot of gas at S.T.P., and the efficiency of the producer, the calorific values of the constituents, in B.Th.U./feet³ at S.T.P. being,

CO	C_2H_4	CH_4	H_2
343	1,678	1,070	296.

Composition of air by volume, O 21 per cent., N 79 per cent.

3. Describe the method of sampling the flue gases of a boiler during a test, and explain, with the aid of sketches of the apparatus used, the method of carrying out the volumetric analysis. State what use is made of this analysis in getting out the heat balance, and indicate the different calculations involved.

4. Define the "efficiency ratio" of a heat engine, and show that its value, for a steam turbine, is given by the ratio of the Rankine engine consumption to the actual consumption. If the isentropic heat-drop in a given turbine is 375 B.Th.U./lbs., and the test consumption at full load is 14 lb./kw. hour at the generator, what is the efficiency ratio of the turbine, the generator efficiency being 95 per cent.? (1 kilowatt = 1·3405 H.P.)

5. Distinguish between the higher and lower calorific values of a gas. Sketch and describe the complete apparatus for the determination of the calorific value of the gas supplied to an engine during a test. State what observations have to be taken, and give the necessary equations for the calculation of the two calorific values, reckoned at standard temperature and pressure (32° F. and 14·7 lb./ins.² abs.).

6. State what is meant by the "critical pressure" in a convergent-divergent steam nozzle. Show that, for initially superheated or dry steam, it is 54·6 per cent. of the initial pressure, the adiabatic exponent being 1·3. Give a reason for the assumption that, within this limit, initially dry steam expands like superheated steam.

7. Fresh water is obtained on board ship from a distilling plant, in which steam evaporated from sea water is condensed in a surface condenser. Dry steam from the evaporator is supplied to the condenser coils at 25 lb./ins.² abs. The circulating pump delivers 13,440 gallons of condensing water per hour. The rise of temperature of the water is 20° F., and the temperature of the condensate is 70° F. Calculate the output of the plant, in gallons of distilled water per day of 24 hours. (Total heat of dry steam at 25 lb./ins.² abs., 1,162 B.Th.U./lbs.).

8. Give a general expression for the heat added to a gas in changing from condition $P_1V_1T_1$ to $P_2V_2T_2$. The compression pressure in a gas engine is 70 lb./ins.² abs., and the temperature 800° F. abs. The maximum pressure, shown by the indicator diagram, after heating at constant volume, is 240 lb./ins.² abs. The gases expand at this pressure, at the beginning of the out-stroke, while the volume increases from 0·2467 to 0·2617 ft.³. Find (*a*) the change of internal energy during the explosion, (*b*) the heat added during the short constant pressure period. Take K = 0·19 and the weight of the gases as 0·056 lb.

9. Dintinguish between "cylinder feed" and "indicated steam," and show how the latter can be calculated from an indicator diagram. State the assumptions made in this calculation. At a point near release on the

expansion curve the pressure is 45 lb./ins abs., and the total volume of steam in the cylinder and clearance spaces is 0·2126 ft.^{3}. At a point on the compression curve the pressure is 25 lb./ins.^{2} abs., and the volume 0·04107 ft.^{3}. The corresponding specific volumes of dry steam at these pressures, are 9·4 and 16·3 ft.^{3}/lbs. The steam consumption of the double-acting engine is 500 lb./hours at 150 revolutions per minute. Find the ratio of indicated steam to cylinder feed.

10. Show from the Tϕ diagram that the efficiency of the constant volume regenerative air engine is the same as that of the Carnot. A Stirling engine, with a perfect regenerator, works between pressure limits of 15 lb./ins.^{2} abs. and 120 lb./ins.^{2} abs., and temperature limits of 60° F. and 580° F. Calculate the thermal efficiency and the compression ratio.

11. State what is meant by the "thermal efficiency" of a steam boiler, and the "factor of evaporation." During a test of an oil-fired boiler, the steam raised at 180 lb./ins.^{2} gauge was 4 per cent. wet. The feed temperature was 60° F., and the evaporation 11 lbs. of water per pound of oil burned. The oil had a calorific value of 19,000 B.Th.U./lbs. Calculate the efficiency and the equivalent evaporation from and at 212° F. Total heat of dry steam at 195 lb./ins.^{2} abs., 1,204·94, latent heat 851·72 B.Th.U./lbs. Latent heat of steam at 212° F. 970 B.Th.U./lbs.

12. A single stage air compressor has a stroke volume V_0 ft.^{3}, and the clearance volume expressed as a percentage of this is c. Air is drawn in at p_1 lb./ins.^{2} abs. compressed to p_2 lb./ins.^{2} abs., and discharged at this pressure. The law of compression and expansion is PV^n = constant. Find an expression for the nett work done on the air during this cycle, and show that it reduces to the standard expression, when clearance is neglected.

April, 1922.

THEORY OF HEAT ENGINES.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. Define the thermal efficiency of a heat engine, and state in the case of the steam engine, how the heat supplied per minute is estimated. During an engine test, steam was supplied at 150 lb./ins.^{2} abs., superheated 120° F., and the temperature at exhaust was 117° F. The weight of condensed steam was 36 lb./mins., and its temperature 107° F. The weight of circulating water was 1,075 lb./mins., and the rise in temperature 35° F. The I.H.P. was 120 and the B.H.P. 108. Calculate the indicated and brake thermal efficiencies, and draw up a tabular heat balance. Total heat of dry steam at 150 lbs./ins.^{2} abs. = 1,199·68 B.Th.U./lbs.; specific heat of superheated steam = 0·558.

2. An air compressor takes in, compresses, and delivers air to a receiver 1 foot diameter and 4 feet long. The initial pressure of the air in the receiver is 15 lb./ins.^{2} abs., and the temperature 60° F. At the end of 45 seconds, the pressure rises to 100 lb./ins.^{2} abs., and the temperature to 120° F. Neglecting leakage loss, calculate the volume of "free air"

per minute (at 60° F. and 14·7 lb./ins.^{2} abs.) taken in by the compressor. Take the gas constant of the characteristic law as 53·3.

3. In getting out the heat balance account for a steam boiler trial, explain, in detail, the methods of calculating the following items :—

(a) Heat transferred to the water.
(b) Total weight of flue gases.
(c) Weight of air for complete combustion.
(d) Weight of excess air.
(e) Weight of products of combustion.

4. The specific heat for gas at constant volume is given by, $K_v = a + sT$, where a and s are constants, and T is the absolute temperature. If W lb. of the gas is heated from T_1 to T_2° F., show that :—

Change of internal energy

$$= W\left[a(T_2 - T_1) + \frac{s}{2}(T_2^2 - T_1^2)\right] \text{ B.Th.U.}$$

During the period of the explosion in a gas engine, the pressure rises at constant volume from 68 lb./ins.^{2} abs. to 180 lb./ins.^{2} abs. The gaseous mixture in the cylinder weighs 0·031 lb., and its temperature, before addition of heat, is 720° F. abs. If 12 B.Th.U. is added during explosion, calculate the heat lost to the jacket water. Take, $a = 0{\cdot}152$ and $s = 0{\cdot}000012$.

5. Describe, with the aid of diagrammatic sketches, the construction and method of operation of a "Uniflow" steam engine, for a large ratio of expansion. Contrast its merits with those of an equivalent multiple expansion engine as regards, (a) steam consumption; (b) temperature range and its effects: (c) clearance losses; (d) piston loads; (e) uniformity of torque. Also explain how excessive compression is prevented in this type of engine.

6. Describe, with the aid of the P V diagram, the cycle of the "open" type of air refrigerating machine. If 1,500 lbs. of air is drawn per hour from the cold chamber at 15 lb./ins.^{2} abs., and 50° F., compressed to 67 lb./ins.^{2} abs., then cooled to 77° F., and expanded again to 15 lb./ins.^{2} abs., calculate the heat taken in and rejected per lb. of air. If the input to the machine is 25 H.P., calculate the actual coefficient of performance. Take the law of compression and expansion as $PV^{1\cdot3} = C$ and specific heat of air at constant pressure as 0·241.

7. State what is meant by the "missing quantity" of a steam engine, and outline the theories advanced to account for it. Indicate the probable effects of the following modifications of the working conditions on the missing quantity, giving a reason in each case : —

(a) Increase of range of expansion.
(b) Steam jacketing of cylinder barrel.
(c) Steam jacketing of cylinder ends and barrel.
(d) Use of superheated steam.
(e) Increase of rotational speed.

8. An engine works on the cycle in which heat is received at constant pressure and rejected at constant volume. Sketch the P V and T φ diagrams for the cycle, and, assuming constant specific heat, show that the efficiency is given by,

$$\eta = 1 - \frac{1}{\gamma r_e^{\gamma - 1}}\left(\frac{\rho^\gamma - 1}{\rho - 1}\right),$$

where r_c is the compression ratio, ρ the ratio of the volume at cut-off to the clearnace volume, and γ the ratio of the specific heats. Contrast the ideal P V diagram with the diagram usually obtained from a Diesel engine.

9. Sketch the combined velocity diagram for a stage of an axial flow reaction steam turbine, assuming equal losses in the fixed and moving ring channels, and from it show that the work done, per lb. of steam, on the moving ring of the pair is given by

$$E = \frac{u^2}{g}\left(\frac{2 \cos \theta}{\rho} - 1\right) \text{ ft.-lb./sec.}$$

where u = mean blade velocity in ft./sec., and $\rho = \frac{\text{blade velocity}}{\text{steam velocity}}$, θ = exit angle of the blade. Show how the diagram is modified in the case of an impulse stage, and give the corresponding expression for the work done.

10. Sketch an arrangement of combined separating and wire-drawing steam calorimeter, connected to the steam pipe of an engine. During a test, the pressure in the pipe was 120 lb./ins.^{2} abs., and the lower pressure at the calorimeter was 16 lb./ins.^{2} abs. The temperature of the steam at this pressure, registered by a thermometer was 268° F. The water collected from the separator was 1·0 lb., and the steam passed through the throttling calorimeter and afterwards condensed, was 24 lbs. Calculate the quality of the steam in the pipe.

Total heat of dry steam at 120 lb./ins.^{2} abs.	= 1,195·06	B.Th.U./lbs.
Latent ,, ,, ,, ,, ,,	= 882·72	,,
Total ,, ,, ,, 16 ,,	= 1,152·48	,,
Temperature ,, ,, ,, ,,	= 216·34° F.	

Specific heat of superheated steam = 0·48.

11. Show by sketches of the Hφ and Tφ diagrams the effect of frictional reheat in a steam turbine nozzle, and find an expression for the actual quality of the steam at exit, when it is finally wet and the reheat is known. If the isentropic heat drop in a nozzle is 300 B.Th.U., and the corresponding (ideal) quality is 0·84. Find the actual quality when the frictional reheat is 15 per cent. of the heat drop. Also calculate the velocity at exit and the weight discharged per hour if the diameter of the nozzle exit is 0·7 inch. (Latent heat of dry steam at 2 lb./ins.^{2} abs. = 1,019·72 B.Th.U./lbs., and specific volume = 173·5 ft.^{3}/lbs.).

12. A two-stage inter-cooled air-compressor works between pressure limits P_1 and P_3. If intercooling is complete, and clearance effect is neglected, show that the work done on the air during suction compression and discharge is a minimum when the suction pressure of the second stage is $P_2 = \sqrt{P_1 P_3}$. Hence find an expression for the total work done on the air. The standard expression for work done in a single stage may be assumed.

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